COMPRESSED AIR PLANT

THE PRODUCTION, TRANSMISSION AND USE OF COMPRESSED AIR

BY

ROBERT PEELE
Mining Engineer and Formerly Professor of Mining in the School of Mines, Columbia University

FIFTH EDITION, LARGELY REWRITTEN

NEW YORK
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ROBERT PEELE, Editor-in-Chief

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PREFACE TO THE FIFTH EDITION

THE first edition of this book was published in 1908. In 1910, and again in 1919, considerable portions were revised, some chapters having been rewritten. Owing to the great changes that have taken place in compressed-air practice in the past decade, it has been necessary in preparing this new edition to rewrite most of the book. The only subjects that have required comparatively little alteration are those dealt with in Chapters I, III, XIV to XVI, XVIII and XXVII of the previous edition.

The following additional statement will enable the reader to trace the principal changes. In this revision the renumbered Chapters, III to V, VII to IX, XI, XII and XIX to XXIII, either consist of entirely new matter or have been rewritten almost throughout; Chapters VI, X, XVI to XVIII and XXIV are condensed and much new material is added.

The structural characteristics of compressors (Chapter III) are exhibited by numerous illustrations, accompanied by outline specifications of the types and sizes made by the principal builders in the United States, Great Britain, France, and in other parts of Europe. Following this are seven chapters dealing with the features of compressor design and operation, including cylinders, valves, governors and unloaders, together with costs and performance of compressors. By means of this rather lengthy presentation of the subject, the author has endeavored to show the trend and achievements of modern practice in a manner that he hopes will be of interest to both builders and users of compressors.

Similar statements may be made respecting the long chapter on Rock-drills. In rewriting this, almost all of the old matter on piston drills has been omitted, as these are now nearly obsolete. They have been superseded by hammer drills for most kinds of rock excavation. Hammer drills have, therefore, been treated at much greater length than in the previous editions of the book. The developments of current practice in their design have been stimulated by keen rivalry among the manufacturers. In connection with the work of rock-drills, the new chapters on Performance of Rock-drills, and Drill Bits and Mechanical Sharpeners, may be found useful.

The author wishes to acknowledge with appreciation the ready assistance received from all the makers of compressed-air plant to whom he has applied for current illustrations and data relating to their machinery. He trusts that the book will meet with their approbation. To Mr. F. R. Schoenborn especial thanks are rendered, for his painstaking and valuable aid.

Robert Peele

New York, March, 1930

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COMPRESSED AIR PLANT

Part First

PRODUCTION OF COMPRESSED AIR

CHAPTER I

INTRODUCTION

ONE of the most important applications of the transmission of power by compressed air is the driving of machine rock-drills; and to the necessity of providing for these drills a power medium suitable for use in mines and tunnels has been due, more than to any other cause, the development of the modern air compressor. A brief résumé of the outstanding facts in the history of air compressors and rock-drills will be interesting to both engineers and students.

The first percussion rock-drill operating independently of gravity was invented in 1849 by J. J. Couch, of Philadelphia. Though used only experimentally, it embodied many of the mechanical features of the modern machine drills. Couch's machine, together with its immediate successors, such as the Fowle drill (1849–51) and the Cavé (Paris, 1851), were steam-driven and therefore unsuitable for underground work. In 1852, the physicist Colladon proposed the use of compressed air for operating rock-drills, in connection with the driving of the Mont Cenis tunnel, in the western Alps. His idea was developed by Sommeiller and others between 1852 and 1860, and in 1861–62 an air-compressor plant was first used successfully at that tunnel. It was driven by water power and furnished air for ventilation as well as for the drills. The early air compressors were crude in design. Sommeiller's first plant, though of large size, had some resemblance in principle to the old hydraulic ram, possessing no moving parts except the valves. Steam-driven piston compressors, as the Dubois-François, and more or less similar to some of the wet compressors still in use in Europe, soon made their appearance. The first compressors built in the United States were the Burleigh, employed at the Hoosac Tunnel, on the Boston and Maine Railroad, in 1865-66. The Norwalk, Clayton, and Rand compressors were among the earlier makes in this country.

In Europe, the Mont Cenis tunnel, about eight miles long, completed in 1871, the first connecting link through the Alps between the railway systems of France and Italy, was the field where were solved on a large scale many problems of compressed-air production and use. Sommeiller there laid the foundations of new practice, by which that great work was successfully completed. From 1857 to 1861 the tunnel headings had been progressing slowly and in the face of great difficulties. Drilling was done by hand labor and blasting by black powder, the average advance for this period, in each of the two headings, being only about 1.5 ft. per day. At this rate, granting that the work could have been finished at all by the means employed, over forty years would have been required to connect the headings and years more to complete the enlargement to full section. With machine drills, the average speed of advance in each heading rose to 4.75 ft. per twenty-four hours and later, when dynamite was introduced, to a little over 6 ft.; this average was maintained for a period of six years.

Machine drills did not make their way into mining to any extent for some years after their successful application to tunnel-driving. It is difficult now to name the mining district in this country where they were first used, but their most important trial was in the Calumet and Hecla copper mine, in Michigan. After strong opposition from the miners, the Rand drill was introduced there in 1878, and the value of machine drilling was soon demonstrated by decreased costs of drifting and stoping and higher speeds of advance.

Compressed air has now a wide application in various branches of mechanical engineering. In this book it is intended to deal chiefly with its production and uses for mining, tunneling and general rock excavation. Its two rivals in these fields of work are steam and electricity.

As compared with steam, compressed-air transmission of power for mines has three important advantages: (1) its loss in transmission through pipes is relatively small; (2) the troublesome question of the disposal of exhaust steam underground is avoided; (3) the exhausted air is of some assistance in ventilating the working places of the mine. In large mines, where steam may be carried thousands of feet, down shafts and through lateral workings, for operating underground machinery, the disadvantages attending its use are apparent; condensation

is serious, even when the piping has good non-conducting covering, and the efficiency becomes abnormally small. Furthermore, aside from the heat produced by the use of steam, it is rarely feasible to employ efficient condensers for underground engines other than pumps, because of the difficulty of obtaining condensing water. If the exhaust is discharged into the mine workings, even though these are large and well ventilated and the volume of the exhaust steam comparatively small, the temperature and quantity of moisture in the air are considerably increased. Deterioration of the timbering is hastened, the roof and walls of the workings are often softened and slaked off, and the mine atmosphere is rendered oppressive and unwholesome. The presence of hot steam pipes in confined workings, or in the narrow compartments of shafts, is also objectionable.

Even with use of the best non-conducting covering, the condensation loss in long steam lines greatly reduces the effective pressure at a distant underground engine and very uneconomical working is the result. In conveying steam several thousand feet the pressure may be reduced to half the boiler pressure, or less. Thus, in the case of a pump situated 2000 ft. from the boiler and using 200 cu.ft. of steam per minute at a boiler pressure of 75 lb., with a 4-in. mineral-wool-covered pipe, the effective pressure at the pump would be only about 58 lb.; or, with a poor covering, like some of the asbestos lagging often used, it might be as low as 35 lb. In compressed-air transmission, on the other hand, the reduction of pressure for the same volume of air, size of pipe, and initial pressure, would be 9.3 lb., giving a terminal pressure of 65.7 lb. However, as the speed of flow in pipes for economical transmission is greater for steam than for air, a comparison based solely on piping of the same diameter cannot justly be made. In the above example, if the diameter of the pipe were smaller the gain in reduced radiation would outweigh the increased frictional loss, and the net loss would be diminished. Since the frictional loss varies inversely, and the loss from radiation directly, with the diameter, the size of the steam pipe can be so proportioned as to produce a minimum loss under given conditions. With compressed air the case is different, since the question of radiation is eliminated. If the pipe diameter is increased to 5 in., the loss of pressure, or head required to overcome friction, is reduced to 2.8 lb., and increasing the distance to one mile would make it only 7.4 lb. Furthermore, as against the increased cost of the larger air pipe, there is the expense of the non-conducting covering necessary for steam transmission.

Thus, compressed air may be conveyed long distances with but small loss of pressure; it is always ready to do its work, and, aside from leakage of pipes, which is preventable, it suffers no loss of power when not in actual use. For performing work *intermittently*, at a distance from its source, it is therefore particularly valuable, because the air pressure is maintained nearly constant during intervals of work, without further expenditure of power. With steam transmission, power is continually dissipated by radiation, and a steam engine, when stopped for any length of time, loses much of its normal working temperature and becomes a receptacle for water of condensation.

Though compressed air is employed in mining mainly for operating machine drills, it is used also for underground hoists and pumps, and sometimes for mechanical coal cutters, in both bituminous and anthracite mines. Compressed-air locomotives in mines and tunnels exemplify its capacity for storing power, in contradistinction to its function as a power transmitter. The introduction of machine drills has facilitated the driving of railroad and mine tunnels, which otherwise would have been greatly delayed or completed only with difficulty. Had compressed-air power, together with the high explosives, not been available, it is doubtful whether the great tunnels in the Alps and elsewhere, and the numerous long mine tunnels driven in recent years in this country, would have been at all practicable.

Without reviewing in detail the comparative merits of electricity and compressed air, it may be pointed out that the application of electricity for transmitting power in mines has increased enormously in importance since about 1888. The peculiar requirements of mine service have been in most cases successfully met by modifications of standard forms of electric apparatus. Both means of power transmission possess characteristics which adapt them particularly for underground work. But, although electricity rivals compressed air in nearly all branches of mine work, their spheres of usefulness are not identical and the field is broad enough for both. Though it is sometimes stated that the first cost of an electric plant is lower than that of an equivalent compressed-air plant, there is actually but little difference between the costs of the plants themselves. For short-distance transmission of a given power an electric conductor line costs much less than compressed-air piping; but the cost of the electric line increases as the square of the distance, whereas that of the pipe line increases only as the first power of the distance. It is in the greater efficiency of generation that electric transmission has the most marked advantage.

In one direction only has electricity failed hitherto to meet every requirement. While compressed-air drills, though far from being economical machines, nevertheless admirably fulfil their purpose, no satisfactory electric rock-drill has yet been produced. It is to be hoped

that a solution of this problem may be found. The Temple-Ingersoll "electric-air" drill, brought out about 1902, is an ingenious machine, but not an electric drill in the ordinary meaning of the term. It is a combination of a compressed-air drill of special design, operated by a small, electric-driven compressor. As there is no exhaust, an incidental advantage of the ordinary air drill is missing, namely, that of assisting somewhat in ventilating those places where ventilation is most needed. This, together with such minor uses of compressed air as the cleaning of drill holes preparatory to charging, and driving out the smoke of blasting from working places, renders it doubtful whether, for underground mining, electric drills of any kind can supersede entirely those operated by compressed air. Given the necessity for a compressed-air plant for rock-drills, as is the case in most metal mines, it may often be more advantageous to provide the additional compressor capacity required for driving underground pumps, hoists, and other machines as well, than to install a separate plant for generating electricity.

It has long been customary to consider compressed air as a mode of power transmission respecting which the questions of convenience and expediency are more weighty than the attainment of a high degree of efficiency. But, as the principles of air compression have become better understood, a substantial improvement has taken place in compressor design, the installation of pipe lines, and the operation of machines using compressed air. The consequences of overloading a compressor, and thereby driving it beyond its proper speed, are comprehended by every intelligent master mechanic as being wholly different from those due to overloading a steam engine. The results of leaks in air pipes, and of using air mains of too small a diameter, are also understood. By better practice in the production, transmission, and use of compressed air a higher total efficiency is now realized than formerly was thought possible.

CHAPTER II

THE COMPRESSION OF AIR

In the production of compressed air occur serious losses, which to a large extent are unavoidable. Even in the best compressors the efficiency, or ratio of the force stored up in the air to the work expended in compressing it, rarely exceeds 75 and often falls below 60 per cent. To understand the causes of these losses it is necessary to study the principles of air compression, and the relations between the pressure, temperature and weight of air. Several definitions follow:

"Free air" is dry air at normal atmospheric pressure and temperature, as taken into the compressor cylinder. Since atmospheric pressure varies with the altitude above sea-level, and with the barometric reading at any particular time or place, the expression "free air" has no precise signification, unless its pressure and temperature are designated. At sea-level it is in reality "compressed air," at the normal atmospheric pressure of 14.7 lb. per sq.in. As commonly employed the term means air at sea-level pressure, and at a temperature of 60° F.

Absolute pressure of air is measured from zero, and is equal to the assumed (or observed) atmospheric pressure plus gage pressure; ordinary gages register pressures in pounds per square inch above atmospheric pressure.

Absolute temperature is the temperature as measured from the "absolute zero" point, which is 491.4° F. below the freezing-point of water, or about 459° below zero F. Thus 60° F. of thermometric temperature is equivalent to an absolute temperature of

$$459° + 60° = 519° F.$$

Two fundamental laws govern the behavior of a perfect gas when undergoing compression, which for practical purposes apply also to atmospheric air. In discussing the problems of air compression, all the relations between volume, pressure, and temperature may be expressed in accordance with these laws. The first law (Boyle's) is:

At constant temperature the volume occupied by a given weight of air varies inversely as the pressure. This is expressed by:

$$PV = P'V' = \text{constant}; \quad \text{or} \quad \frac{P'}{P} = \frac{V}{V'};$$

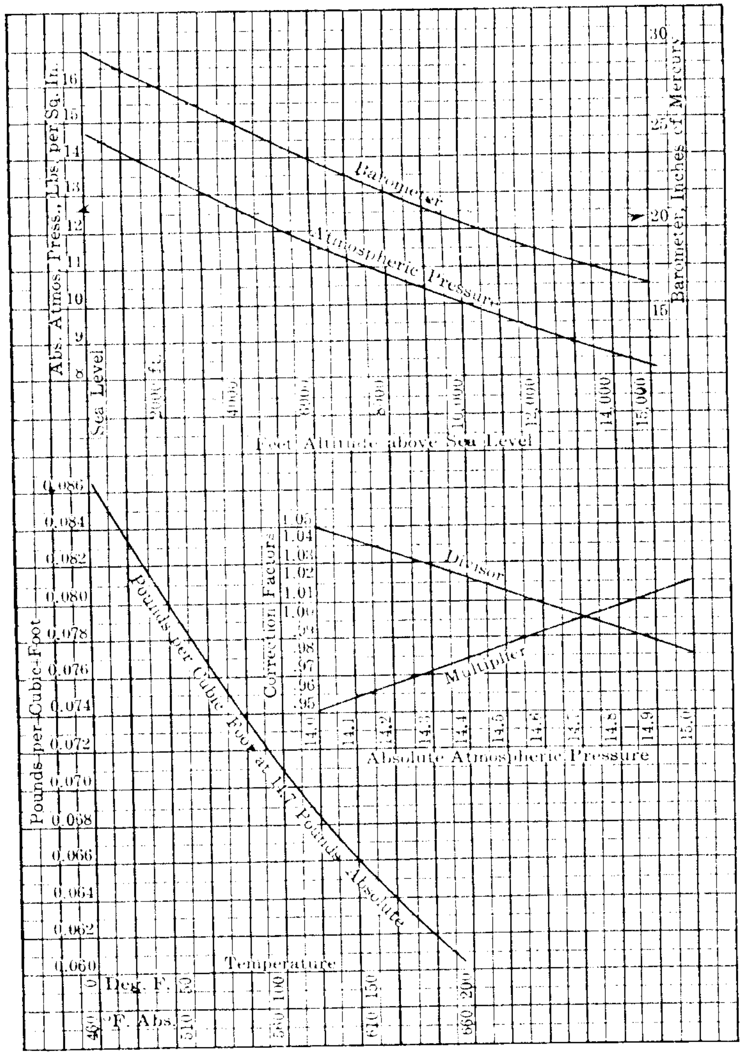

FIG 1.—Pressure, Temperature and Weight Relations of Air (Chas. W. Gibbs; from "Compressed Air Data").

in which V = volume of the given weight of air (or gas) at the freezing-point and at a pressure P (V usually being taken as the volume in cubic feet occupied by 1 lb. of air);

V' = volume of the same weight of air at the same temperature and at any pressure P' (the pressures being absolute pressures).

For example, to compress a quantity of atmospheric air at constant temperature to 0.147 of its original volume (atmospheric pressure being 14.7 lb.), requires a pressure of 100 lb. per sq. in.; when compressed to 0.074 of its original volume, the pressure required is 200 lb., and so on.

TABLE I. WEIGHT AND VOLUME OF DRY AIR AT SEA-LEVEL PRESSURE AND DIFFERENT TEMPERATURES

Temperature, deg. F.	Weight of 1 cu.ft., in lb.	Volume of 1 lb., in cu.ft.	Temperature, deg. F.	Weight of 1 cu.ft., in lb.	Volume of 1 lb., in cu.ft.
0	.0864	11.582	110	.0698	14.345
10	.0846	11.834	120	.0686	14.596
20	.0828	12.085	130	.0674	14.847
30	.0811	12.336	140	.0663	15.098
32	.0807	12.386	150	.0652	15.350
40	.0795	12.587	160	.0641	15.601
50	.0780	12.838	170	.0631	15.852
60	.0764	13.089	180	.0621	16.103
62	.0761	13.141	190	.0612	16.354
70	.0750	13.340	200	.0603	16.605
80	.0736	13.592	210	.0593	16.856
90	.0723	13.813	212	.0591	16.907
100	.0710	14.094			

The production and use of compressed air are not governed solely by Boyle's law. During compression heat is generated, and when the air is allowed to re-expand to its original volume this heat is taken up. If there is no transference of heat, the internal work, manifested by the increase of temperature, is independent of the time occupied by compression. This condition is expressed by the second law, that of Charles and Gay-Lussac: When under constant pressure, the volume of a gas expands or contracts for each degree rise or fall of temperature, from freezing to boiling, by a constant fraction of the volume which is occupied at the freezing-point. Stated in another way, the volume of a gas under constant pressure is nearly proportional to the absolute temperature. The equation may be written: $V' = V(1 + at°)$. The complete relations between pressure, volume, and temperature are expressed by the equation: $P'V' = PV(1 + at°)$, in which P' and V' represent

TABLE II.—WEIGHT OF AIR AT VARIOUS TEMPERATURES AND PRESSURES

(Sea-level pressure = 14.7 lb. abs.)

Temperature of Air, Deg. F.	Gage Pressure, Lb.																	
	0	5	10	20	30	40	50	60	70	80	90	100	110	120	130	140	150	175	200
	Weight, Lb. per Cu.ft.																		
−10	.0882	.1184	.1485	.2090	.2685	.3283	.3880	.4478	.5076	.5674	.6272	.687	.747	.807	.868	.928	.989	1.139	1.288
0	.0864	.1160	.1455	.2040	.2630	.3215	.3800	.4385	.4970	.5555	.6140	.672	.731	.790	.849	.908	.968	1.114	1.260
10	.0846	.1136	.1425	.1995	.2568	.3145	.3720	.4292	.4863	.5433	.6006	.658	.716	.774	.832	.889	.947	1.090	1.233
20	.0828	.1112	.1395	.1955	.2516	.3071	.3645	.4205	.4770	.5330	.5890	.645	.701	.757	.813	.869	.927	1.067	1.208
30	.0811	.1088	.1366	.1916	.2465	.3015	.3570	.4121	.4672	.5221	.5771	.632	.687	.742	.797	.852	.908	1.046	1.184
40	.0795	.1067	.1338	.1876	.2415	.2954	.3503	.4038	.4576	.5114	.5652	.619	.673	.727	.781	.835	.890	1.025	1.161
50	.0780	.1045	.1310	.1839	.2367	.2905	.3432	.3960	.4487	.5014	.5541	.607	.660	.713	.766	.819	.873	1.006	1.139
60	.0764	.1025	.1283	.1803	.2323	.2840	.3562	.3882	.4402	.4927	.5447	.596	.649	.700	.752	.804	.856	.988	1.116
70	.0750	.1005	.1260	.1770	.2280	.2791	.3302	.3808	.4316	.4824	.5332	.584	.635	.686	.737	.788	.839	.967	1.095
80	.0736	.0988	.1239	.1738	.2237	.2739	.3242	.3738	.4234	.4729	.5224	.572	.622	.673	.723	.774	.824	.949	1.074
90	.0723	.0970	.1218	.1707	.2195	.2688	.3182	.3670	.4154	.4639	.5122	.561	.611	.660	.709	.759	.809	.932	1.054
100	.0710	.0954	.1197	.1676	.2155	.2638	.3122	.3602	.4079	.4555	.5033	.551	.599	.648	.696	.745	.794	.914	1.035
110	.0698	.0937	.1176	.1645	.2115	.2593	.3070	.3542	.4011	.4481	.4950	.542	.589	.637	.685	.732	.780	.899	1.017
120	.0686	.0921	.1155	.1618	.2080	.2549	.3018	.3481	.3944	.4403	.4866	.533	.579	.626	.673	.720	.767	.884	1.001
130	.0674	.0905	.1135	.1590	.2045	.2505	.2966	.3446	.3924	.4296	.4770	.524	.570	.616	.662	.708	.754	.869	.984
140	.0663	.0889	.1115	.1565	.2015	.2465	.2915	.3364	.3813	.4262	.4711	.516	.561	.606	.651	.696	.742	.855	.968
150	.0652	.0874	.1096	.1541	.1985	.2425	.2865	.3308	.3751	.4193	.4636	.508	.552	.596	.640	.685	.730	.841	.953
175	.0626	.0840	.1054	.1482	.1910	.2335	.2755	.3181	.3607	.4033	.4450	.488	.531	.573	.616	.658	.701	.808	.914
200	.0603	.0809	.1014	.1427	.1840	.2248	.2655	.3054	.3473	.3882	.4291	.470	.511	.552	.592	.633	.674	.776	.879
225	.0581	.0779	.0976	.1373	.1770	.2163	.2555	.2949	.3344	.3738	.4129	.452	.491	.531	.570	.609	.649	.747	.846
250	.0560	.0751	.0941	.1323	.1705	.2085	.2466	.2845	.3223	.3602	.3981	.436	.474	.513	.551	.589	.627	.722	.817
300	.0523	.0707	.0881	.1237	.1592	.1945	.2300	.2654	.3008	.3362	.3716	.407	.442	.478	.513	.549	.585	.673	.762
350	.0491	.0658	.0825	.1160	.1495	.1828	.2160	.2492	.2824	.3156	.3488	.382	.415	.449	.482	.516	.549	.632	.715
400	.0463	.0621	.0779	.1090	.1405	.1720	.2035	.2348	.2661	.2974	.3287	.360	.391	.423	.454	.486	.517	.596	.674
450	.0437	.0586	.0735	.1033	.1330	.1628	.1925	.2220	.2515	.2810	.3105	.340	.369	.399	.429	.458	.488	.562	.637
500	.0414	.0555	.0696	.0978	.1260	.1540	.1820	.2100	.2380	.2660	.2940	.322	.351	.379	.407	.435	.463	.534	.604

the pressure and volume of a given weight of air (or gas) at $t°$ F. above the freezing-point, V is the volume of the same quantity at the freezing-point, and a the coefficient of expansion of air, which is practically constant and is very nearly $\frac{1}{491}$ on the Fahrenheit scale. Hence, for a rise in temperature of 1° F., the volume of air increases by $\frac{1}{491}$ of the volume occupied at the freezing-point, under the same pressure (491° F. being the absolute temperature below freezing).

The practical application of this law is that the heat generated reacts upon the air under compression, and increases the pressure due merely to the reduction in volume. By cooling the compressed air to its original temperature the pressure would be reduced to the normal amount, according to the first law. That is, the heat produced by compressing a given volume of air corresponds in degree to the cold resulting from the re-expansion of the same quantity of air to its original volume and pressure.

Two other statements may be deduced from what precedes: (1) Under constant pressure the volume of air varies directly as the absolute temperature; (2) The volume being constant, the absolute pressure varies directly as the absolute temperature.

The first of these statements is expressed thus:

$$\frac{V}{t} = \frac{V'}{t'} = \text{constant},$$

in which t and t' are absolute temperatures; whence, from Boyle's law:

$$P\frac{V}{t} = P'\frac{V'}{t'} = \text{constant}.$$

This constant is denoted by R, the general equation being $\frac{PV}{t} = R$.

The value of R is found as follows: Since for a given weight of gas or air the density, D, is inversely proportionate to the volume, $V = \frac{1}{0.08073}$, 0.08073 being the weight, in pounds, of 1 cu.ft. of dry air, at sea-level pressure (14.7 lb.) and 32° F. The normal atmospheric pressure per square foot = 14.7 × 144 = 2116.8 lb. If, therefore, 1 cu.ft. is expanded by the application of heat to a volume of 2 cu.ft., the work done against atmospheric pressure, per pound of air, will be $\frac{2116.8 \times 1}{0.08073} = 26{,}220$ ft.-lb. To double the volume, according to Charles' law, would require the expenditure of 491.4° F. of heat;

whence, in raising the temperature 1° F., the external work done by expansion is:

$$\frac{PV}{t} = \frac{26220}{491.4} = 53.37 = R,$$

and R equals foot-pounds of work done by expanding under constant pressure one pound of air, while enough heat is added to increase its temperature 1° F.

The heat generated during compression and corresponding to different pressures is shown in Table III, the volume at normal atmospheric pressure being 1, at a temperature of 60° F. This table shows that the *rate* of increase of temperature is not uniform, but diminishes as the pressure rises. Thus, from 1 to 2 atmospheres the increase is 115.8°; from 2 to 3, 79.3°; from 3 to 4 atmospheres, 62.3°, etc. The quantity of heat generated during compression may be calculated by the following formula:

TABLE III. HEAT GENERATED DURING COMPRESSION OF AIR

Pressure in atmospheres	Absolute pressures, lb. per sq. in. above vacuum	Volumes in cu.ft., adiabatic compression	Final temperatures, deg. F.	Corresponding increases of temperature
1.00	14.70	1.000	60.0	00.0
1.25	18.37	0.854	94.8	34.8
1.50	22.05	0.750	124.9	64.9
2.00	29.40	0.612	175.8	115.8
2.50	36.70	0.522	218.3	158.3
3.00	44.10	0.459	255.1	195.1
3.50	51.40	0.411	287.8	227.8
4.00	58.80	0.374	317.4	257.4
5.00	73.50	0.319	369.4	309.4
6.00	88.20	0.281	414.5	351.5
7.00	102.90	0.252	454.5	394.5
8.00	117.60	0.229	490.6	430.6
9.00	132.30	0.211	523.7	463.4
10.00	147.00	0.195	554.0	494.0
15.00	220.50	0.147	681.0	621.0
16	[illegible]		[illegible]	

$$Q = \frac{R \times t}{J} \times \text{Nap. log}\ \frac{V'}{V},$$

in which Q = quantity of heat in thermal units;

R = constant = 96.037 (French unit) or 53.37 (English unit);

t = absolute final temperature in degrees, corresponding to V' (centigrade scale for French and Fahrenheit for English units);

J = value of one thermal unit = 1400 ft. lb. (or 778 ft. lb. if English units are used). (Joule's equivalent.)

V and V' = volumes of air, in cubic feet, at beginning and end of compression.

As the rise in temperature due to compression is proportional to the ratio of the final absolute pressure to the initial absolute pressure, the quantity of heat generated during compression to any given pressure, and the consequent work done per unit volume of compressed air, are greater at altitudes than at sea-level.

The above conclusions are illustrated by the diagram, Fig. 2.* It is, in reality, two diagrams, combined to save space. First, beginning at the lower left-hand corner, and curving upward, are the adiabatic and isothermal compression lines. Their intersections with the horizontal and vertical lines give the volumes of the unit of air when subjected to any given pressure, by reading the figures at the top, and right- or left-hand margin of the diagram. The initial volume is taken as 1, and the spaces between the vertical lines are each one-tenth. The resulting volume is independent of the initial temperature of the air. The corresponding pressure may be read in terms of either gage or atmospheric pressure. Second, beginning at the lower right-hand corner of the diagram, and rising toward the left, are the lines of temperature, the assumed initial temperatures being 0°, 60° and 100° F. The temperature corresponding to any given pressure is read on the lower margin. It should be observed that these temperature curves are those of adiabatic compression.

It follows from the above that if the temperature of the air rises during compression an increase of work ensues.

Isothermal and Adiabatic Compression. In accordance with the laws already stated, air may be compressed in two ways:

Isothermal Compression.—The temperature is kept constant during compression, the heat generated being abstracted as fast as it is produced. In this case the pressure of the air varies according to the equation $PV = P'V'$, and the compression curve of an indicator diagram is an isothermal curve.

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* Taken from "Compressed Air Production," by W. L. Saunders, several slight corrections having been made in the adiabatic and isothermal lines.

Adiabatic Compression.—The temperature may be allowed to rise unchecked during the period of compression, as it will when there is no transference of heat by radiation or cooling devices. The rise in temperature increases the pressure due to reduction of volume only. Thus, the pressure rises faster than the volume diminishes, and $\frac{P'}{P} > \frac{V}{V'}$. This relation is determined as follows. The specific heat of a gas or vapor at constant pressure, C_p, is the quantity of heat required to raise the temperature of 1 lb. of the gas 1° F., the pressure being unchanged. The specific heat at constant volume, C_v, is the quantity of heat required to raise the temperature of the gas 1° F., the volume being unchanged. The specific heat of moist air is computed by considering it as a mixture of air and steam. For dry air, $C_p = 0.2375$ and $C_v = 0.1689$. C_p is the greater, because external work is done during a change of temperature, if the pressure is constant and the air free to expand; under constant volume, no work is done upon external resistances. When, as in adiabatic compression the heat generated reacts on the air under compression and increases the value of $\frac{P'}{P}$, to maintain the equation $\frac{V}{V'}$, must be increased by an amount equivalent

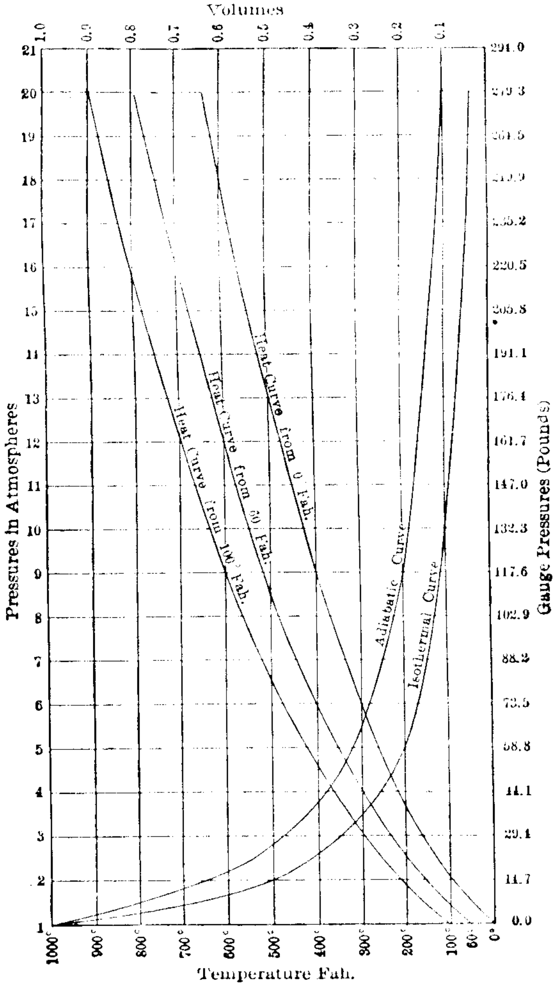

FIG. 2.

to the external work performed. The specific heats may be expressed in heat units, as above; or, by multiplying them by the mechanical equivalent of a heat unit (778 ft.-lb. = J), they are given in terms of foot-pounds and are then denoted by K; that is,

$$JC_p = K_p \quad \text{and} \quad JC_v = K_v.$$

Since $K_p = C_p \times 778 = 184.77$, and $K_v = C_v \times 778 = 131.4$:

$$\frac{K_p}{K_v} = \frac{184.77}{131.4} = 1.406.$$

This ratio is commonly denoted by n, and is the exponent of the power to which $\frac{V}{V'}$ must be raised to make it equal to $\frac{P'}{P}$ *; n may also be expressed as equal to the ratio of the specific heats at constant pressure and volume:

$$\frac{C_p}{C_v} = \frac{0.2375}{0.1689} = 1.406 = n.$$

The general equation for adiabatic compression is therefore:

$$PV^n = P'V'^n \quad \text{or} \quad \frac{P'}{P} = \left(\frac{V}{V'}\right)^{n = 1.406}$$

Work of Compressors without Clearance. Isothermal Compression. The work done by a compressor without clearance, and using isothermal compression, is represented by the area under the compression curve (Fig. 3). Let AB be an isothermal curve, AD representing any volume V of free air, and BC the volume V', to which this quantity of air is compressed; the corresponding absolute pressures being respectively P and P' The curve is an equilateral hyperbola, and the work W represented by the area

$$ABCD = W_1 + W_2 - W_3,$$

in which W_1 = area under $AB = \int PdV$;

W_2 = area under $BC = P'V'$, representing the work of expelling the air from the cylinder;

W_3 = area under $DA = PV$, representing the negative work done by atmospheric pressure on the suction or intake side of the piston.

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* A statement of the proof of this deduction is unnecessary here; it is given in several books on Thermodynamics, for example, in Perry's work on the Steam Engine.

Since $PV = P'V'$, W_2 and W_3 cancel, so that the algebraic sum of

$$W = W_1 + W_2 - W_3 = \int PdV. \tag{1}$$

To integrate this expression, substitute for P its equivalent $\frac{P'V'}{V}$:

$$W = \int_{V'}^{V} P'V' \frac{dV}{V} = P'V' \int_{V'}^{V} \frac{dV}{V}$$

Integrating:

$$W = P'V' \times \text{Nap. log.} \left(\frac{V}{V'}\right) * \tag{2}$$

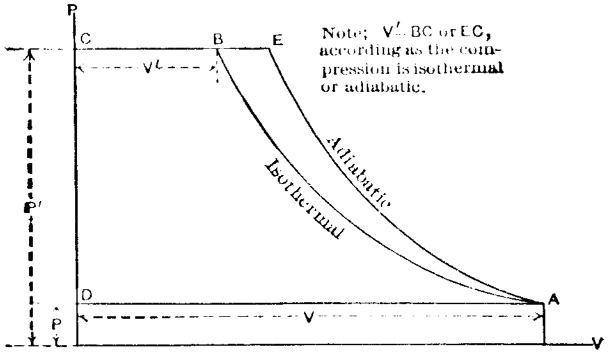
P
C
B
E
Note; V' BC or EC, according as the compression is isothermal or adiabatic.
V'
P'
Adiabatic
Isothermal
D
A
P
V
V

Fig. 3.—Reference Diagram.

The equation may also be written:

$$W = PV \log_e \left(\frac{P'}{P}\right) \tag{3}$$

a form convenient for use in making air compressor calculations. When expressed in foot-pounds (by putting V in terms of cubic feet, and P, P' in pounds per square inch):

$$W = 144PV \log_e \left(\frac{P'}{P}\right) \tag{4}$$

which is the general equation for the work of compressors operating isothermally and without clearance.

Work of Compressors without Clearance. Adiabatic Compression. Referring to Fig. 3, the line AD represents the initial volume,

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* The Napierian or hyperbolic logarithm of a number, generally written "$\log_e$," is obtained by multiplying the *common* logarithm by the constant 2.302585.

V, of air at normal atmospheric pressure, and the line EC the final volume V', to which the same quantity of air is *compressed*; that is, V' is the volume after the compression part of the stroke is completed and before delivery begins. In undergoing this change of volume, the pressure increases *from* P *to* P', and the resulting compression line AE is an adiabatic curve, following the law:

$$PV^n = P'V'^n = C \text{ (constant)}, \quad \text{or} \quad P = \frac{C}{V^n} \tag{5}$$

The total work done in the compressing cylinder is:

$$W = (W_1 + W_2 - W_3) \tag{6}$$

in which W_1 = the work of compression;
W_2 = work required to force the compressed air out of the *cylinder, into the receiver*;
W_3 = work done by atmospheric pressure on the suction side of the piston, while the inlet air is entering the cylinder.

(1) *Work of Compression.*—The work W_1, in foot-pounds, is:

$$W_1 = \int_{V'}^{V} 144PdV$$

Substituting the value of P, now expressed in pounds per square inch, from equation (5)

$$W_1 = 144\int_{V'}^{V} \frac{CdV}{V^n} \tag{7}$$

Integrating between the limits V and V':

$$W_1 = 144C\left[\frac{V^{(1-n)} - V'^{(1-n)}}{1-n}\right] \tag{8}$$

Dividing *the second member of the equation* by (-1) and substituting for C its value PV^n:

$$W_1 = \frac{144PV^n}{n-1}\left[\frac{1}{V'^{(n-1)}} - \frac{1}{V^{(n-1)}}\right] \tag{9}$$

$$= \frac{144PV}{n-1}\left[\frac{V^{(n-1)}}{V'^{(n-1)}} - 1\right] \tag{10}$$

But, since $\dfrac{P'}{P} = \dfrac{V^n}{V'^n}, \dfrac{V}{V'} = \left(\dfrac{P'}{P}\right)^{\frac{1}{n}}$, which, raised to the $n - 1$ power, gives:

$$\frac{V^{(n-1)}}{V'^{(n-1)}} = \left(\frac{P'}{P}\right)^{\frac{n-1}{n}} \tag{11}$$

Substituting this value in (10):

$$W_1 = \frac{144PV}{n-1}\left[\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} - 1\right] \tag{12}$$

(2) *Work of Expulsion.*—The work of expelling the air from the cylinder is,

$$W_2 = 144P'V' \tag{13}$$

Multiplying by $\dfrac{V'}{V}$ both members of the expression, $\dfrac{P'}{P} = \left(\dfrac{V}{V'}\right)^n$:

$$\frac{P'V'}{PV} = \left(\frac{V}{V'}\right)^{n-1}; \quad \text{whence,} \quad P'V' = PV\left(\frac{V}{V'}\right)^{n-1}.$$

But,

$$\frac{V}{V'} = \left(\frac{P'}{P}\right)^{\frac{1}{n}} \quad \text{and} \quad \left(\frac{V}{V'}\right)^{n-1} = \left(\frac{P'}{P}\right)^{\frac{n-1}{n}};$$

hence $P'V' = PV\left(\dfrac{P'}{P}\right)^{\frac{n-1}{n}}$; which, substituted in equation (13), gives:

$$W_2 = 144P'V' = 144PV\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} \tag{14}$$

(3) *Work done by Atmospheric Pressure.*—The work W_3, done by atmospheric pressure on the back of the piston, $= 144PV$. (15)

Taking the algebraic sum of W_1, W_2 and W_3, from equations (12), (14) and (15), and substituting in equation (6):

$$W = 144\left\{\frac{PV}{n-1}\left[\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} - 1\right] + PV\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} - PV\right\};$$

whence, by reducing to a common denominator:

$$W = 144\frac{PV\left[\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} - 1\right] + (n-1)PV\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} - (n-1)PV}{n-1}$$

and canceling:

$$\mathbf{W = \frac{144PVn}{n-1}\left[\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} - 1\right]} \tag{16}$$

which is the general expression for the work of single-stage compressors, with adiabatic compression, and when clearance is zero.

The relations between the two conditions of compression are represented graphically by Fig. 4. By laying off to scale the volumes of air on the horizontal line of the diagram, the corresponding pressures at different points of the stroke of the piston are measured on the verticals. The adiabatic curve rises more rapidly than the isothermal, meaning that more work is expended. Perfect isothermal compression is unattainable. It is only approximated even with the best cooling arrangements, and running the compressor at a very slow speed. On the other hand, if the air compressed adiabatically could be kept hot until used, the loss of the additional work which was expended in compressing it would be prevented. But neither can this be done. The air is usually conveyed considerable distances before it is used, and radiation from the pipes soon reduces the pressure to that corresponding with the temperature of the surrounding atmosphere. In practice, a combination of the two theoretical modes of compression is employed, the net result depending upon the degree of perfection of the compressor and of the cooling arrangements. When compressing in a single cylinder to 60 or 80 lb., and a piston speed not exceeding 300 ft. per min., it is probable that about one-half of the total possible cooling is all that may be expected.* The aim is to begin compression with the air at a low initial temperature, and to bring the compression line as close as possible to the isothermal line. The air must be cooled thoroughly during compression and before it leaves the cylinder; any subsequent cooling, in the receiver or in the air main, entails loss.

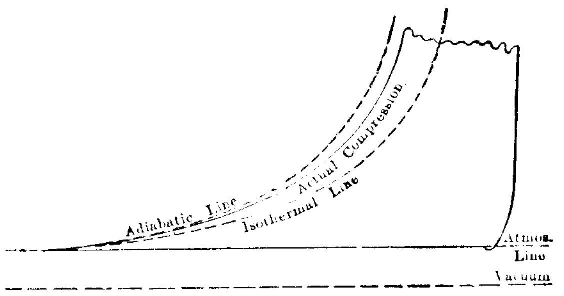

Fig. 4.

In ordinary practice the abstraction of heat during compression is very imperfect. Some distance must be traversed by the compressing piston before there is any considerable rise in temperature, and until the temperature does rise no cooling can be effected. The abstraction of heat does not begin at the beginning of the stroke. The temperatures of the intake air and of the cooling water are likely to be nearly the same, so that all the possible reduction of temperature in any one cylinderful

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* Frank Richards, "Compressed Air."

of air must take place in a period of time less than that occupied in making the stroke. In modern dry compressors of fairly large size, and running at full working speed, the compression line is usually much nearer the adiabatic than the isothermal curve, and often follows the adiabatic curve quite closely.

The heat produced by compression may be absorbed:

1. By introducing cold water into the air cylinder.
2. By cooling the cylinder from without, enveloping it in a cold-water jacket.

Machines of the first class are known as "wet compressors," now obsolete; those of the second, "dry compressors."

Values of *n* in the equation $\frac{P'}{P} = \left(\frac{V}{V''}\right)^n$. In purely adiabatic compression, $n = 1.406$; in ordinary single-cylinder dry compressors, n is roughly 1.3, whereas in the best single-stage wet compressors (with spray injection) n becomes 1.2 to 1.25. In the poorest forms of compressor the value $n = 1.4$ is closely approached. For large, well-designed stage compressors and efficient intercooling, n, referred to the combined indicator cards (Fig. 7), may be as small as 1.15.

Work of Two-stage Compressors, without Clearance. The air is brought up to a certain pressure in one cylinder; passes to an intercooler, in which the temperature of the air is reduced; and finally enters a second cylinder, where the compression is carried to the desired terminal pressure. The cylinder ratio is such that the work is equally divided between the cylinders, but changes of conditions of operation other than those contemplated may destroy this equality. In the following discussion * it is assumed that the same quantity of work is done in both stages.

An inspection of the diagram, Fig. 5, shows that there must be some best intermediate receiver pressure, for which the total work of compression will be a minimum. For, if this receiver pressure approaches either P or P'' (corresponding respectively to the points B and G on the compression curve), then would the compression approach single-stage work and the entire compression line would lie along BCG. But, with an intercooling receiver at any intermediate point, the broken line $BCDE$ is followed, the saving in work over single-stage compression being represented by the area $\overline{CDEG}$.

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* The author desires here to acknowledge the valuable assistance of Dr. Charles E. Lucke, Professor of Mechanical Engineering in Columbia University, who kindly gave him the use of his original notes on the method of analysis employed in this discussion.

The net work of the compressor, W, represented by the area $\overline{ABCDEF}$, is equal to the work of area $\overline{ABCH}$ of the first stage plus the work of area $\overline{HDEF}$ of the second stage, or $W = W_1 + W_2$. Let the condition of perfect intercooling back to the air temperature corresponding to the isothermal line be assumed; that is, the hot air discharged from the first cylinder is cooled in the intermediate receiver to the initial temperature of the intake air. The work cycle in each cylinder is the same as that of single-stage adiabatic compression, as expressed by equation (16), but with two additional symbols for pressures and volumes.

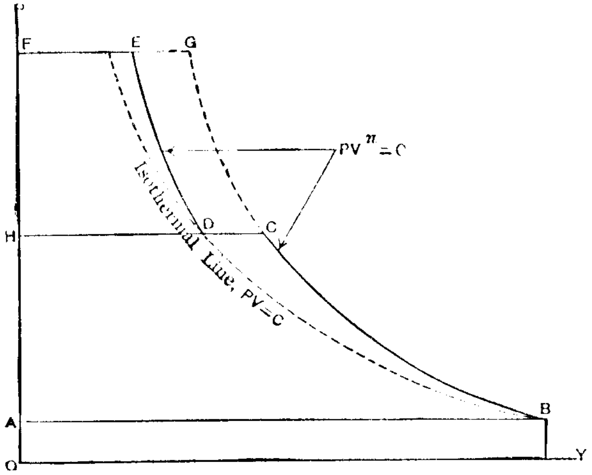

FIG. 5.—Reference Diagram, Two-Stage Compressor, with no Clearance and Perfect Intercooling.

Let $AB = V$ = initial volume of free air in first cylinder;
$HD = V''$ = initial volume of air in second cylinder;
$OA = P$ = initial absolute pressure (atmospheric pressure);
$OH = P'$ = terminal absolute pressure in first cylinder, assumed to be also the intermediate receiver pressure and therefore the initial pressure in the second cylinder;
$OF = P''$ = terminal absolute pressure in second cylinder.

Hence:

$$\left.\begin{aligned} W_1 &= \frac{PVn}{n-1}\left[\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} - 1\right] \quad \text{first stage} \\ W_2 &= \frac{P'V''n}{n-1}\left[\left(\frac{P''}{P'}\right)^{\frac{n-1}{n}} - 1\right] \quad \text{second stage} \end{aligned}\right\} \qquad (17)$$

But, assuming the intercooling to be perfect, $PV = P'V''$, whence:

$$W = \frac{PVn}{n-1}\left[\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} + \left(\frac{P''}{P'}\right)^{\frac{n-1}{n}} - 2\right] \qquad (18)$$

Since the best receiver pressure, P', is that for which W is a minimum, by differentiating and placing the first differential coefficient $\frac{dW}{dP'} = 0$:

$$\frac{dW}{dP'} = PV\frac{n}{n-1}\left\{\frac{n-1}{n}\frac{(P')^{\frac{n-1}{n}-1}}{P^{\frac{n-1}{n}}} - \frac{n-1}{n}\frac{(P'')^{\frac{n-1}{n}}}{(P')^{\frac{n-1}{n}+1}}\right\} = 0$$

Whence:

$$\frac{(P')^{-\frac{1}{n}}}{P^{\frac{n-1}{n}}} = \frac{(P'')^{\frac{n-1}{n}}}{(P')^{\frac{2n-1}{n}}}$$

or

$$(P')^{-\frac{1}{n}+\frac{2n-1}{n}} = (PP'')^{\frac{n-1}{n}};$$

from which is derived an expression for the best receiver pressure:

$$P' = \sqrt{PP''}.$$

Dividing both terms by P:

$$\frac{P'}{P} = \frac{(P \times P'')^{\frac{1}{2}}}{P} = \left(\frac{P''}{P}\right)^{\frac{1}{2}}.$$

But, $\left(\frac{P''}{P}\right)^{\frac{1}{2}} = \frac{P''}{\sqrt{PP''}} = \frac{P''}{P'}$. Substituting these values in equation (18), remembering that $P'V'' = PV$ and expressing the work in foot-pounds:

$$W = \frac{2 \times 144PVn}{n-1}\left[\left(\frac{P''}{P}\right)^{\frac{n-1}{2n}} - 1\right] \qquad (19)$$

which is the equation for two-stage compressor work, in terms of the initial volume and initial and terminal pressures, with perfect intercooling and best receiver pressure.

By a method similar to the above, the expression for the work of three-stage compression may also be deduced:

$$W = \frac{3 \times 144PVn}{n-1}\left[\left(\frac{P'''}{P}\right)^{\frac{n-1}{3n}} - 1\right] \qquad (20)$$

in which P''' is the terminal pressure in the last, or high-pressure, cylinder.

Effect of Clearance in the Compressing Cylinder. In the preceding pages expressions are deduced for the work of compression with no allowance for the clearance volume of the cylinder. From a mechanical engineering and structural point of view, the question of piston clearance is discussed in Chapter V. It is necessary here to discuss the cycles of operation of single-stage and two-stage compression with clearance. Although the work done per unit of air is the same as that shown by the general equations for isothermal and adiabatic compression, the work per unit of cylinder displacement will be changed, because of the re-expansion of the clearance air. In other words, clearance affects the volumetric output of the compressor, but not the work of compression per unit volume of air taken into the cylinder.

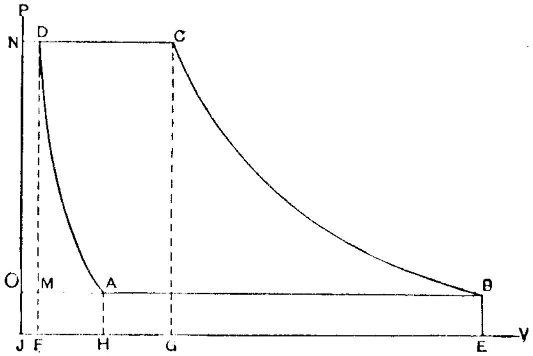

Fig. 6.—Reference Diagram. Compressor Working Isothermally, with Clearance.

Work of Compressors with Clearance. Isothermal Compression. Fig. 6 is a general reference diagram, in which BC represents an isothermal curve.

Let $EB = MF = P$;
$GC = FD = P'$;
$JE = OB = V$;
$CN = V'$;
$JF = ND$ = clearance volume;
$JH = OA = (OB - AB) = V'''$, or volume occupied by the re-expanded clearance air.

According to the diagram areas:

Net work $A\overline{BCD}$ = compression work and delivery work $OBCN$ − re-expansion work $OADN$.

For the work of a compressor without clearance, and isothermal compression, the general expression, $W = PV \log_e \left(\frac{P'}{P}\right)$, has already been deduced. This applies to the areas bounded by the two horizontal lines, the vertical line and the compression line. Similarly, the re-expansion work represented by the area

$$OADN \text{ (under the curve } DA) = PV''' \log_e \left(\frac{P'}{P}\right). \quad \text{Hence,}$$

$$W = PV \log_e \left(\frac{P'}{P}\right) - PV''' \log_e \left(\frac{P'}{P}\right) = P(V - V''') \log_e \left(\frac{P'}{P}\right) \quad (21)$$

Replacing $(V - V''')$ by L, which represents the intake capacity of the compressing cylinder, neglecting heating during suction, and expressing P in pounds per square inch:

$$W = 144PL \log_e \left(\frac{P'}{P}\right) \quad (22)$$

Comparing equations (4) and (22) it is seen that they are identical, as noted above; but it must be remembered that the volume of air actually taken into the cylinder at each stroke and compressed is reduced on account of clearance, and hence the volumetric capacity of the compressor is also reduced. Moreover, neither in this work cycle, nor in that for adiabatic compression, is any account taken of the heating and cooling effects which occur during intake and compression, nor of frictional and other losses which affect capacity and work per units of free and compressed air. These points are discussed elsewhere in this chapter and in Chapters V, X and XII.

Single-stage Adiabatic Compression, with Clearance. The diagram, Fig. 6, may be used here also, by assuming the line BC to be an adiabatic curve. But, though the work areas are designated as above, under isothermal compression, and their significations are identical, their numerical values are different.

From equation (16), the work corresponding to area $\overline{OBCN}$ is

$$W_1 = \frac{144PVn}{n-1}\left[\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} - 1\right];$$

similarly, the work corresponding to the area $\overline{OADN}$ is

$$W_2 = \frac{144PV'''n}{n-1}\left[\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} - 1\right].$$

Whence,

$$W = \frac{144P(V - V''')n}{n-1}\left[\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} - 1\right] \quad (23)$$

Replacing $(V - V''')$ by L (the intake capacity of the cylinder, with clearance):

$$W = \frac{144PLn}{n-1}\left[\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} - 1\right] \qquad (24)$$

Since V in equation (16) may be replaced by L, a comparison of this equation with (24) shows that the work is the same per unit of volume of air admitted to the cylinder; but, the volumetric output is reduced by the clearance.

Though the pressure-volume formulas serve for most purposes, it is sometimes convenient to have the work expressed in terms of cylinder displacement and volumetric efficiency:

Referring to Fig. 6:

Let D = displacement volume of cylinder in cubic feet, or the effective area × stroke, represented on the diagram by $MB = FE$;

C = clearance expressed as a fraction of D; whence $C \times D = V_c$ = clearance volume, represented by $ND = JF$, and $D(1 + C)$ = total cylinder volume in cubic feet, represented by JE;

V''' = volume of re-expanded clearance air;

L = intake free air capacity $= JE - JH = V - V'''$;

E = volumetric efficiency $= \frac{L}{D}$ = the ratio of the length of the actual admission line, AB, to the total distance swept through by the piston.

Then $V = D(1 + C)$, and $V''' = V_c\left(\frac{P'}{P}\right)^{\frac{1}{n}} = CD\left(\frac{P'}{P}\right)^{\frac{1}{n}}$.

Whence $\qquad V - V''' = D\left[1 + C - C\left(\frac{P'}{P}\right)^{\frac{1}{n}}\right] = L.$

Substituting this value of $V - V'''$ in equation (23):

$$W = \frac{144Pn}{n-1}D\left[1 + C - C\left(\frac{P'}{P}\right)^{\frac{1}{n}}\right]\left[\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} - 1\right] \qquad (25)$$

which expresses the work in terms of displacement and clearance, with pressures in lb. per sq. in.

Since $E = \frac{L}{D}$, $L = ED = D\left[1 + C - C\left(\frac{P'}{P}\right)^{\frac{1}{n}}\right]$, as above; whence, substituting in (25):

$$W = \frac{144Pn}{n-1}ED\left[\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} - 1\right]. \qquad (26)$$

Work of Stage Compressors, with Clearance (Fig. 7). The method of analysis of the work of a two-stage compressor, without clearance (equation 19), applies also here. With clearance, the work for each stage is expressed by equation (23), the necessary changes being made in the signification of the different factors:

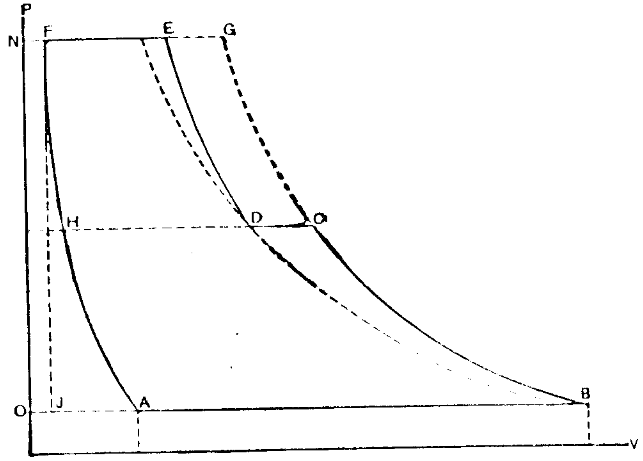

Fig. 7.—Reference Diagram. Work of Two-Stage Compressor, with Clearance.

$$\left.\begin{aligned} W_1 \text{ (first stage)} &= \frac{144P(V - V''')n}{n-1}\left[\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} - 1\right] \\ W_2 \text{ (second stage)} &= \frac{144P'(V'' - V^{\text{IV}})n}{n-1}\left[\left(\frac{P''}{P'}\right)^{\frac{n-1}{n}} - 1\right] \end{aligned}\right\} \qquad (27)$$

In equation (27) the symbols have the same significations as in equations 18, 24, in addition to which V^{IV} is the volume of re-expanded clearance air of the high-pressure cylinder. But, if intercooling is perfect,

$$PV - PV''' = P'V'' - P'V^{\text{IV}},$$

that is, the weight of air entering the second cylinder is equal to that entering the first. Hence, the general equation for the work of a two-stage compressor, with clearance and perfect intercooling, takes the form:

$$W = \frac{144P(V - V''')n}{n-1}\left[\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} + \left(\frac{P''}{P'}\right)^{\frac{n-1}{n}} - 2\right] \qquad (28)$$

By differentiating this equation, and making the first differential coefficient $\frac{dW}{dP'} = 0$, it is found that $P' = \sqrt{PP''}$. This is the value for the best (most economical) intermediate receiver pressure. It is the same as that deduced in equations 19, 20 for stage compression without clearance, since the receiver pressure is a function of the compression line and not of the re-expansion line.* By substituting in equation (27) the value of P' for best receiver pressure, it will be found that, as in the work cycle for stage compression without clearance, there is here also an equal division of work between the two cylinders. Finally, if the same value of P' is substituted in equation (28), this equation takes the form:

$$W = \frac{2 \times 144P(V - V''')n}{n-1}\left[\left(\frac{P''}{P}\right)^{\frac{n-1}{2n}} - 1\right] \qquad (29)$$

The diagram (Fig. 7) shows a single, continuous re-expansion line, *FA*, which is taken to represent the re-expansion lines of both cylinders. This evidently is true only when the clearances are proportionate; that is, when the *volume of the clearance air* of the high-pressure cylinder, after re-expansion, is equal to the *clearance volume* of the intake cylinder. But the cylinders of stage compressors may, and usually do, have different clearances, between which no particular relation exists. However, since the quantity of air delivered by the low-pressure cylinder must necessarily be equal to that received by the high-pressure cylinder, disproportionate clearance does not affect the work done per unit of air compressed. The diagram of the high-pressure is merely displaced somewhat with respect to that of the low-pressure cylinder, as shown by Fig. 8, in which $FE = F'E'$ and $HD = H'D'$.

In a manner similar to the above may be deduced the expression for

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* For the sake of brevity, the steps in the deduction of these and the following work formulas are omitted. Readers who desire to pursue the subject further will find a full discussion in Lucke's *Engineering Thermodynamics*.

work of a three-stage compressor, with clearance and best intermediate receiver pressures:

$$W = \frac{3 \times 144P(V - V''')n}{n-1}\left[\left(\frac{P'''}{P}\right)^{\frac{n-1}{3n}} - 1\right], \qquad (30)$$

in which P''' is the delivery pressure, and V''' the re-expansion volume of the intake cylinder.

The work and capacity of stage compressors may also be expressed in terms of displacement and clearance.

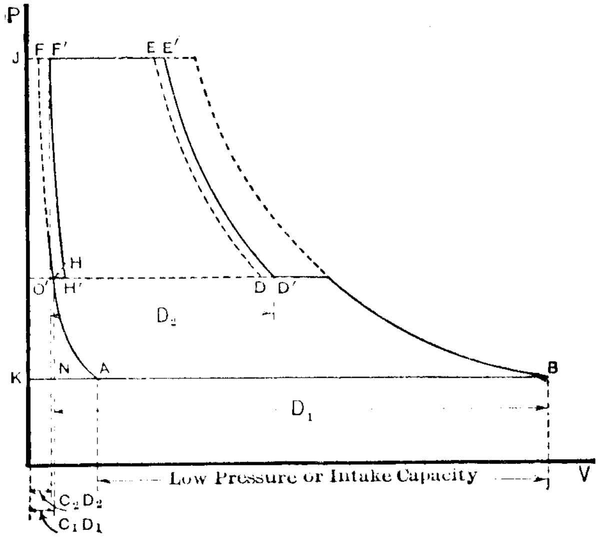

FIG. 8.—Reference Diagram. Work of Two-Stage Compressor, with Disproportionate Clearance.

Let D_1 and D_2 = cylinder displacements, respectively of the low- and high-pressure cylinders;

C_1 and C_2 = fractional clearances; whence $C_1 \times D_1$ and $C_2 \times D_2$ = clearance volumes, and $D_1(1 + C_1)$ and $D_2(1 + C_2)$ = total cylinder volumes, all in cubic feet.

$E_1 = \frac{AB}{NB}$ = volumetric efficiency of low-pressure cylinder;

$E_2 = \frac{H'D'}{O'D'}$ = volumetric efficiency of high-pressure cylinder.

In the demonstration leading to equation (25) it was found that

$$V = V''' = D\left[1 + C - C\left(\frac{P'}{P}\right)^{\frac{1}{n}}\right].$$

Applying the proper subscripts for two-stage work, and substituting this value of $V - V'''$ in equation (28), remembering that the volume discharged by the low-pressure is equal to that received by the high-pressure cylinder:

$$W = \frac{144Pn}{n-1}D_1\left[1 + C - C_1\left(\frac{P'}{P}\right)^{\frac{1}{n}}\right]\left[\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} + \left(\frac{P''}{P'}\right)^{\frac{n-1}{n}} - 2\right] \quad (31)$$

which expresses the work in terms of displacement, clearance, and initial and terminal pressures.

Since, when the intercooling is perfect, the temperature of the air entering the high-pressure cylinder is the same as that entering the intake cylinder, it follows that the product of the initial pressure in each cylinder multiplied by the volume of air admitted to each is the same for both cylinders. As previously stated, $P(V - V''') = P'(V'' - V^{\text{iv}})$, and, $P' = (PP'')^{\frac{1}{2}}$ for best receiver pressure. Therefore,

$$P(V - V''') = (PP'')^{\frac{1}{2}}(V'' - V^{\text{iv}});$$

whence

$$\frac{V - V'''}{V'' - V^{\text{iv}}} = \frac{(PP'')^{\frac{1}{2}}}{P} = \left(\frac{P''}{P}\right)^{\frac{1}{2}}.$$

But, since $V - V''' = D_1E_1$ and $V'' - V^{\text{iv}} = D_2E_2$,

$$\left(\frac{P''}{P}\right)^{\frac{1}{2}} = \frac{D_1E_1}{D_2E_2}. \quad (32)$$

According to the reasoning which led to equation (26),

$$D_1E_1 = D_1\left[1 + C_1 - C_1\left(\frac{P'}{P}\right)^{\frac{1}{n}}\right] \quad (33)$$

and

$$D_2E_2 = D_2\left[1 + C_2 - C_2\left(\frac{P''}{P'}\right)^{\frac{1}{n}}\right] \quad (34)$$

Dividing (33) by (34) and combining with (32):

$$\left(\frac{P''}{P}\right)^{\frac{1}{2}} = \frac{D_1E_1}{D_2E_2} = \frac{D_1}{D_2}\,\frac{\left[1 + C_1 - C_1\left(\frac{P'}{P}\right)^{\frac{1}{n}}\right]}{\left[1 + C_2 - C_2\left(\frac{P''}{P'}\right)^{\frac{1}{n}}\right]}. \quad (35)$$

For convenience in applying this equation, it may take the form:

$$\frac{D_1}{D_2} = \left(\frac{P''}{P}\right)^{\frac{1}{2}}\frac{\left[1 + C_2 - C_2\left(\frac{P''}{P}\right)^{\frac{1}{2n}}\right]}{\left[1 + C_1 - C_1\left(\frac{P''}{P}\right)^{\frac{1}{2n}}\right]} \quad (36)$$

This transformation follows from the relation, for best receiver pressure:

$$\left(\frac{P'}{P}\right)^{\frac{1}{n}} = \left(\frac{P''}{P'}\right)^{\frac{1}{n}} = \left(\frac{P''}{P}\right)^{\frac{1}{2n}}.$$

Equation (36) expresses the ratio between the displacements of the cylinders, in terms of initial and terminal pressures and clearances.

If the percentage clearances of the cylinders are equal, $C_1 = C_2$ and $E_1 = E_2$; whence, the quantities in brackets of the second member of equation (36) cancel, and $\frac{D_1}{D_2} = \left(\frac{P''}{P}\right)^{\frac{1}{2}}$. The same would be true if C_1 and $C_2 = 0$; a condition which may be assumed for compressors having very small clearance.

From equations (35) and (36), for the condition of best receiver pressure, several relations may be determined: (1) The ratio of compression for a given ratio of cylinder capacities, or conversely; (2) the ratio of cylinder displacements for known volumetric efficiencies; (3) the ratios of compression in the two cylinders which will produce best receiver pressure, the displacements and clearances being known, or conversely. In the third case, several approximations will usually be required.

For the performance of air compressors see Chapter X. Tables are there given, showing the work actually required per cubic foot of free air, for single-, two- and three-stage compression.

CHAPTER III

STRUCTURAL CHARACTERISTICS OF COMPRESSORS

An air compressor consists of a cylinder in which atmospheric air is compressed by a piston, the driving power being derived from a steam engine or other motor. Except for the high-pressure cylinders of multi-stage compressors, the air cylinder is double-acting, with inlet and delivery valves in each head. When the pressure in the cylinder, behind the advancing piston, falls below atmospheric pressure the inlet valves open automatically, due to this difference in pressure, and air flows into the cylinder. In a few designs the valves are under partial mechanical control, to aid their prompt opening and closure. At each stroke of the piston a certain volume of compressed air is forced out through the discharge valves, into a pipe leading to a receiver, whence the air enters the transmission pipe.

CLASSIFICATIONS

Compressors are broadly divisible into two principal classes: direct *steam-driven* and *power-driven*, the latter including (*a*) those in which the air cylinders are operated by belt or gearing from electric motors, oil engines or water-wheels, and (*b*) those in which the air cylinders are direct-connected to the power element.

Subdivisions of each of these two principal classes are given in the following tabulation, which comprises compressors for nearly all kinds of service. Certain special designs of compressors of small capacity, like those for operating air brakes and vacuum pumps, which differ from the others only in their mechanical features, are not included, as this book deals mainly with plant applicable to the great field of mining, tunneling and general rock excavation.

Steam-Driven

Straight-line	Single-stage, double acting	Direct-connected to steam engine, or with long- or short-belt drive
	Single stage, double-acting, portable	Direct-connected to engine
Duplex, small and medium capacity	Single-, two- or three-stage	Direct-connected to engine, or with long- or short-belt drive
Duplex, large capacity	Single- or two-stage	Direct-connected to engine
Turbo-compressors		Direct-connected to turbine

Power-Driven

Vertical	Single-stage, single-acting, stationary	For long belting to line shaft, oil engine or electric motor For short-belt drive
	Duplex single-stage, single-acting, portable	For short-belt drive
Straight-line	Single-stage, double-acting	For long- or short-belt drive For chain drive (rarely used)
	Single-stage, double-acting, portable	For short-belt drive For chain drive (rarely used)
Duplex, small and medium capacity	Single-, two- or three-stage	Direct-connected to electric motor or oil engine, or with long- or short-belt drive
Duplex, large capacity	Single- or two-stage	Direct-connected to electric motor, gas or oil engine, or water-wheel For long-belt drive

The three-stage compressors as tabulated are suitable for charging compressed-air locomotives. Four-stage compressors built by several makers, comprise gas compressors and those employed in some of the chemical industries and other special service. Illustrations and specifications of multi-stage compressors are given in this chapter.

Following is a more detailed classification, based also on the *structural characteristics* of compressors. Features of design having to do with their *thermodynamic operation*, *i.e.*, the modes of minimizing the heat produced in the air cylinders during the act of compression, and with their air valves, speed and pressure regulators, etc., are dealt with in Chapters II, V to X and XII.

1. Straight-line Compressors.—The cylinders are horizontal and placed tandem on a common piston-rod. There may be a single flywheel, or a pair of flywheels, one on each end of the crank-shaft, driven by a single connecting-rod (Figs. 1, 2, 3, 5, 8, 9, 11, 14 and 15); or, rarely, by outside connecting-rods from a crosshead between the cylinders. The compressor may be single-stage, with one steam and one air cylinder, or two-stage air and compound steam cylinders; or, it may be driven by belt or direct-connected motor.

2. Duplex Compressors.—Two engines are placed side by side, each consisting of tandem steam and air cylinders, with their cranks at 90° on a common flywheel shaft. Each side of the duplex is in effect a straight-line compressor. The steam cylinders are simple, tandem-compound or cross-compound; the air cylinders, single or staged (Figs. 10, 16 to 20, 25 and 26). Considerations leading to compounding the steam end generally make it desirable to use stage compression. Nearly all large steam-driven compressors are now duplex, cross-compound, two-stage, and are of the horizontal, vertical, or angle type (see Class **4**).

Classes **1** and **2** are made by all the principal compressor builders in the United States; class **2**, also by some British builders.

3. Vertical Compressors: (*a*) power-driven, by belt or direct-connected electric motor (Figs. 6 and 17); (*b*) steam-driven, the air cylinders being set above the steam cylinders (Figs. 22, 24 and 35). The air end is simple or staged.

These have had their largest development in Great Britain, including the compressors of W. H. Bailey & Co., Belliss & Morcom, Peter Brotherhood & Co., Reavell & Co., Robey & Co. and Walker Brothers. In the United States, vertical three- and four-stage compressors are built by the Norwalk Co., and small to medium size two-stage compressors by Allis-Chalmers Co., Chicago Pneumatic Tool Co., Ingersoll-Rand Co., Sullivan Machinery Co., and Worthington Pump & Machinery Co.

4. Angle Compressors, represented chiefly by several designs of the Sullivan Machinery Co., have two-stage air cylinders set at 90°. The low-pressure cylinder is horizontal, the high-pressure vertical, with a common crank-shaft, driven by belt, direct-connected electric motor or Diesel engine (Fig. 33). For large capacities they are of the twin form.

The Worthington Co. makes a medium-size angle compressor, having a horizontal, two-stage compressing unit, driven by a vertical gas or oil engine. Type POV-2 of the Ingersoll-Rand Co. (Fig. 28) has a vertical, differential air cylinder, driven by a horizontal oil engine. These two designs, however, are not true "angle"-compressors according to the class definition.

5. Turbo and Rotary Compressors.—The turbo type (Figs. 47 and 48), originally designed for low air pressures only, is now built to produce, in a number of stages, pressures suitable for mine service. Turbos of large capacity are made by Adamson & Co., British Thomson-Houston Co., British Westinghouse Electric Co., Fraser & Chalmers Engineering Works, General Electric Co., Ingersoll-Rand Co., and several other makers.

Rotary compressors, of relatively small sizes, are built by Holman Bros., Reavell & Co. and others, chiefly for service other than mining (Figs. 12, 32 and 50).

For a list of the principal compressor builders in the United States, Great Britain and the continent of Europe, see end of this chapter.

On the following pages are given tabulated sizes and specifications of standard compressors of different makers. As these specifications and accompanying illustrations are presented here to exhibit the general range of practice in compressor design, it will be understood that the

author has no desire to discriminate, nor to express a preference for any particular make. In a book that is not a trade publication, it is neither necessary nor practicable to include examples of all builders. The examples chosen are arranged in alphabetical order, and are to be considered chiefly as representative of compressors suitable for mine service and general rock excavation. Compressors of a number of makers in Great Britain and several on the continent of Europe are cited, as it is believed that these designs, many of which differ essentially from those of American builders, will be interesting and suggestive to all concerned with compressed air practice.

This chapter deals only with the different makes, types and sizes of compressors, from the general mechanical engineering standpoint. Details of the air and steam cylinders, including the valves and their operation, compressor regulators, together with the comparison and performance of compressors, are set forth in Chapters **IV** to **VIII** and X.

Choice of Compressor Plant.—The selection of a compressor for any given service involves detailed study of all the local conditions. It is a problem of engineering economics, the chief factors of which are: first cost of plant, transport facilities, estimated fuel or power consumption, cost of fuel or electric current, maintenance charges, cost of attendance, and overhead charges (including depreciation, interest, etc.). As most of these factors can be expressed in dollars and cents per unit of operating time, a clear comparison of the different types and makes of plant may be drawn, leading directly to the choice of a suitable installation. The proper interpretation of the results of such a study presupposes both engineering training and experience; and the advice of reputable compressor builders is always desirable.

Compressors Having Simple Air Cylinders

With a few exceptions, as noted, simple or single-stage compressors are of the horizontal, "straight-line" form (see preceding classifications). They may be driven by steam, gasoline or fuel oil-engines, by direct-connected electric motor, or by belt. In general, they are of small to medium size, that is, having piston displacements up to 1,700 or 1,800 cu. ft. of free air per minute, and producing gage pressures of 30 to 125 lb. per sq. in. (sometimes lower pressures for particular service).

American Air Compressor Works. Straight-line, Steam-driven, Class WS (Fig. 1)

Cylinder diam., in.		Stroke, in.	Rev. per min.	Displacement, cu.ft. per min.	Air pressure, lb.	I. Hp.		Air pipe, in.		Floor space, in.	
Steam	Air					Min.	Max.	Inlet	Disch.	Length	Width
7	6¼	6	350	74	80 125	9	11	2	2	87	26
7	7¼	6	350	100	65 100	11	14	2½	2	87	26
9	8¼	8	300	148	80 125	21	26	2½	2½	114	34
9	9¼	8	300	186	65 100	24	30	3	3	114	34
9	10¼	8	300	230	35 60	24	31	3	3	114	34
12	10¼	10	275	273	80 125	40	48	3½	3	135	42
12	12¼	10	275	375	60 100	52	64	4	3½	135	42
14	12¼	12	250	410	80 125	63	73	4	3½	164	48
14	14¼	12	250	555	45 100	69	94	5	4	164	48

Note. Thirteen other sizes of this class are built for air pressures of 10 to 55 lb. per sq. in.

The American Air Compressor Works have a line of twenty-eight belt-driven, single-stage compressors, with cylinders from 4¼ by 4 in. to 20¼ by 12 in., and displacements of 31 to 1,100 cu.ft. per min.; also, three small sizes of vertical compressors, with maximum displacement of 61 cu.ft. per min.

W. H. Bailey & Co. Straight-line, Steam-driven, Class **B** (Fig. 2)

For pressures of 45 to 60 lb. per sq.in.

Cylinder diam., in.		Stroke	Rev. per min.	Free air del'd per min., cu.ft.	Brake hp. for air press. of		Size of air pipe, in.	
Steam	Air				45 lb.	60 lb.	Inlet	Disch.
6.3	5.8	5.8	250	40	6.25	7.5	2	2
7.8	7.8	7.1	230	80	11 13	14	2.5	2.5
8.8	9.8	9.8	200	150	23	28	3.5	3.5
9.8	10.8	11.8	175	188	29	33 35	4	4
10.8	11.8	13.8	160	250	39	43 46	4.5	4.5
13.8	14.75	13.8	160	400	59 63	71 75	6	6
14.8	16.75	15.8	150	530	75 78	90 95	7	6
15.8	17.75	17.75	150	700	98 103	120 125	8	6
16	20.7	17.75	150	950	130 140	160 170	9	7
18	24.0	17.75	150	1,275	160 170	200 215	9	7

Note.—Belt-driven compressors, Class A, have air cylinders of the same dimensions, with same speed of stroke.

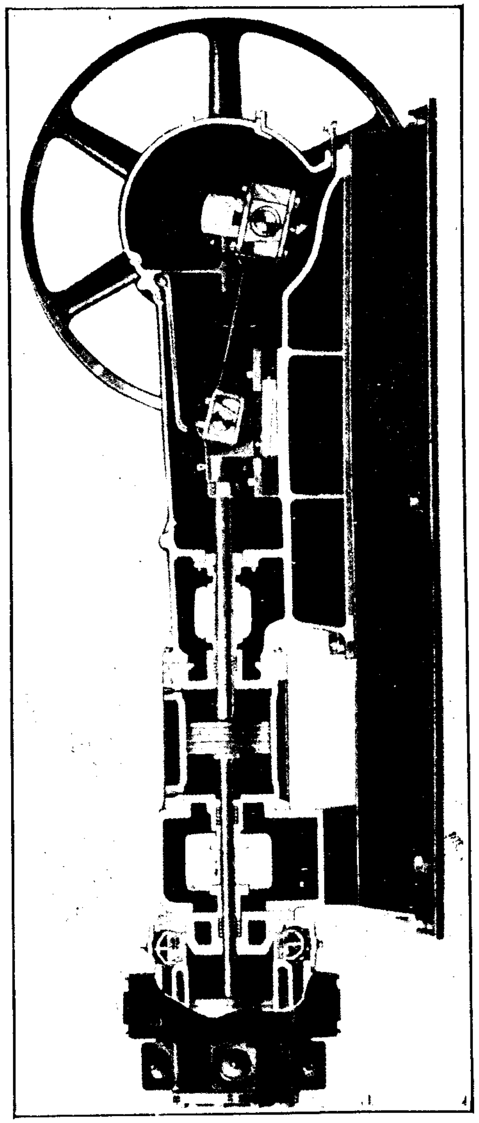

FIG. 1.—American Air Compressor Works. Straight-line, Single-stage Compressor, Class WS.

The design of Bailey & Co.'s compressors differs from typical American practice in having piston valves for both steam and air cylinders (see Chapter IV, Fig. 4, and Chapter VII, Figs. 5 and 6).

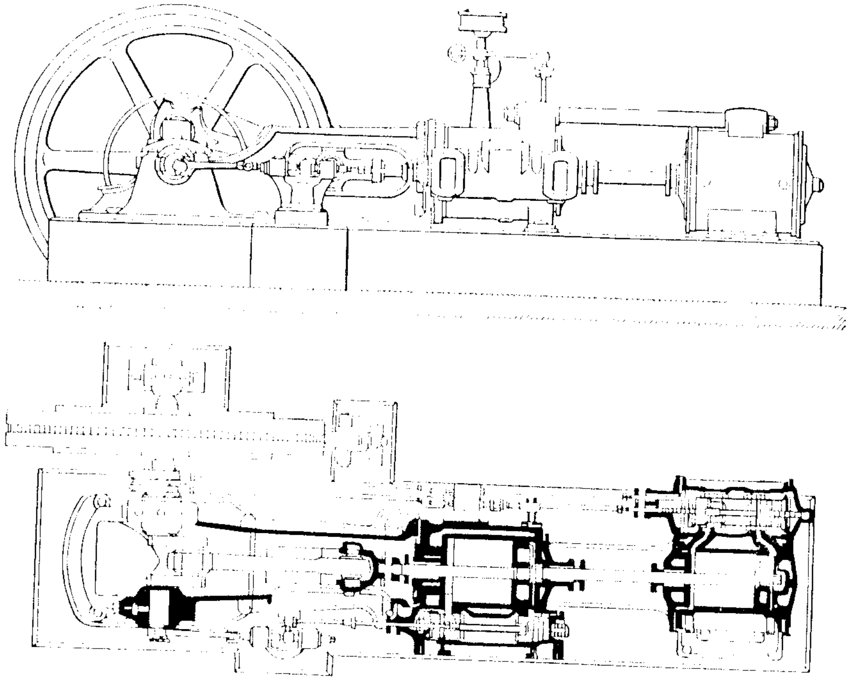

FIG. 2.—W. H. Bailey & Co. Straight-line, Steam-driven Compressor, Class B.

Chicago Pneumatic Tool Co. Straight-line, Steam-driven, Class N-SS (Fig. 3)

Cylinder diam., in.			Rev. per min.	Dis-placement, cu.ft. per min.	Max air press., lb.	Size of air pipes, in.		Floor space			
Steam	Air	Stroke				Inlet	Disch.	Length, ft.	in.	Width, ft.	in.
6	6	6	350	67	125	2	2	7	4	2	6
6	7½	6	350	106	100	2½	2	7	4	2	6
6	9	6	350	153	50	3	3	7	8	2	6
8	8	8	300	136	125	3	3	9	3	3	0
8	9	8	300	173	100	3	3	9	3	3	0
8	10	8	300	215	70	3	3	9	3	3	0
8	12	8	300	311	40	4	4	9	3	3	0
10	10	10	275	216	125	4	4	10	5	3	5
10	12	10	275	356	100	4	4	10	5	3	5
10	15	10	275	558	50	5	5	10	7	3	5
10	17	10	275	719	50	6	6	11	2	3	5
13½	12	12	250	387	125	5	5	12	3	3	9
13½	14	12	250	529	100	5	5	12	3	3	9
13½	17	12	250	783	50	6	6	12	6	3	9
13½	20	12	250	1,085	30	8	8	13	3	3	9
15	17	14	235	857	80	6	6	13	10	4	7
15	20	14	235	1,189	40	8	8	14	2	4	7

Notes. The steam cylinders of the 6-, 8- and 10-in. stroke "N" Class compressors have "D" slide-valves; 12-in. stroke, double-port balanced valves. The air cylinders have "simplate" disk-valves.

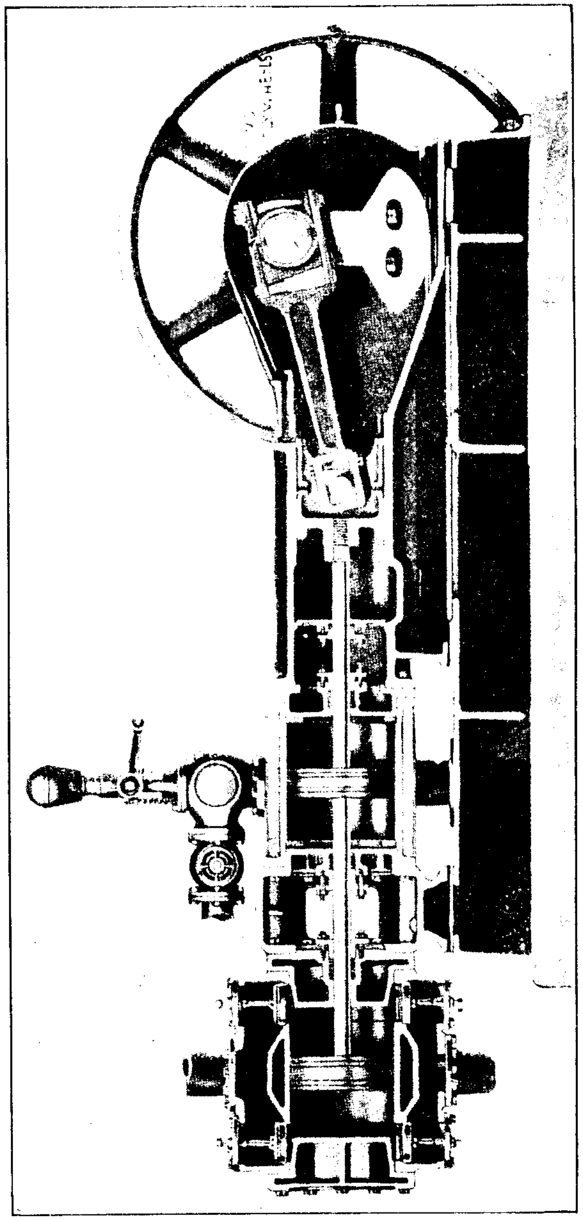

FIG. 3.—Chicago Pneumatic Tool Co. Straight-line, Single-stage Compressor, Class N-SS.

The Chicago Pneum. Tool Co. also builds a number of sizes of duplex, single-stage compressors, Class "O," driven by steam, belt or direct-connected electric motor, for pressures of 30 to 70 lb., and displacements from 1,022 to 4,124 cu.ft. per min.

Chicago Pneumatic Tool Co. Belt-driven, Class N–SB (Fig. 4)

Cylinders, in.		Rev. per min.	Displacement, cu.ft. per min.	Max. air press., lb. per sq. in.	Diam. air pipes, in.		Floor space	
Diam.	Stroke				Inlet	Disch.	Length, ft. in.	Width, ft. in.
6	6	350	67	125	2	2	5 10	2 2
7½	6	350	106	100	2½	2	5 10	2 2
9	6	350	153	50	3	3	6 2	2 2
8	8	300	136	125	3	3	7 4	3 1
9	8	300	173	100	3	3	7 4	3 1
10	8	300	215	70	3	3	7 4	3 1
10	10	275	246	125	4	4	8 2	3 10
12	10	275	356	100	4	4	8 2	3 10
15	10	275	558	50	5	5	8 4	3 10
12	12	250	387	125	5	5	9 7	4 2
14	12	250	529	100	5	5	9 7	4 2
17	12	250	783	50	6	6	10 0	4 2
17	14	235	857	80	6	6	11 6	5 1

Note.—Four other sizes of this class are built for air pressures of 30 to 40 lb.

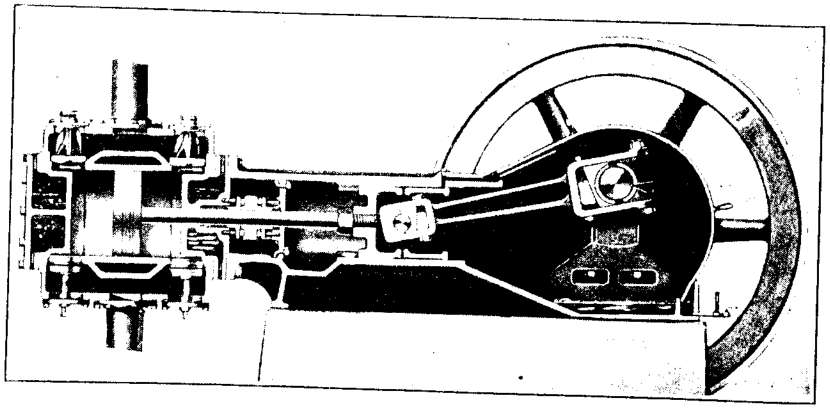

Fig. 4.—Chicago Pneumatic Tool Co. Single-stage, Belt-driven Compressor, Class N SB.

Chicago Pneumatic Tool Co. Fuel Oil-driven, Class N-SO2 (Fig. 5)

Air pressures to 100 lb. per sq. in.

Displacement, cu.ft. per min.	Rev. per min.	Max. Hp.	Air pipes, in.		Water supply pipes, in.		Floor space	
			Inlet	Disch.	Air cyl.	Oil cyl.	Length, ft. in.	Width, ft. in.
144	325	21	5	3½	¾	1¼	11 9	3 5
212	325	29	5	4	¾	1¼	13 6	3 8
309	300	47	6	6	¾	1½	15 4	4 7
370	250	57	6	6	¾	1½	18 3	5 7
507	230	77	8	8	¾	2	21 6	6 6
658	230	102	8	8	¾	2	21 10	6 6

Dujardin & Cie. Horizontal, Belt-driven, Type PH

Air pressure, 57 lb. per sq.in.

	Cylinders, in.*		Rev. per min.	Piston displacement	Actual delivery	Hp.	Approx. floor space, in.	
	Diam.	Stroke		Cu.ft. per min.			Length	Width
Single cyl.	$9\frac{7}{16}$	$9\frac{7}{8}$	350	280	194	34	84	37
	$11\frac{3}{8}$	$11\frac{13}{16}$	325	452	318	55	102	43
	$13\frac{3}{4}$	$13\frac{13}{16}$	300	710	508	85	120	52
	$15\frac{3}{4}$	$15\frac{3}{4}$	275	978	705	116	132	57
	$17\frac{3}{4}$	$17\frac{3}{4}$	250	1,270	920	149	150	63
	$20\frac{1}{4}$	$19\frac{11}{16}$	231	1,700	1,265	204	168	67
Duplex	$15\frac{3}{4}$	$15\frac{3}{4}$	275	1,956	1,410	232	132	96
	$17\frac{3}{4}$	$17\frac{3}{4}$	250	2,540	1,840	298	150	102
	$20\frac{1}{4}$	$19\frac{11}{16}$	231	3,400	2,530	408	168	114

* Cylinder dimensions computed to nearest sixteenth inch from metric scale.

Dujardin & Cie. also build lines of small vertical, single-cylinder compressors, Types PW and PV, and large vertical, duplex compressors, Types CS and BS.

Holman Bros. Duplex, Vertical, Double-acting, Belt or Rope-driven, Type B.X. 1 (Fig. 6)

Cylinders, in.		Rev. per min.	Displacement, cu.ft. per min.	Hp. at crankshaft		Air pipe, in.		Floorspace, in.	
Diam.	Stroke			For 60 lb.	For 80 lb.	Inlet	Disch.	Length	Width
9	7	380	385	58	68	4½	3	81	46
11½	8	340	645	96	110	5	4	86	48
13	9	320	870	130	154	7	4½	100	57
15½	10	300	1,290	192	228	8	5½	110	58

Approximate weights of above compressors: 8,500, 13,000, 15,100 and 19,500 lb.

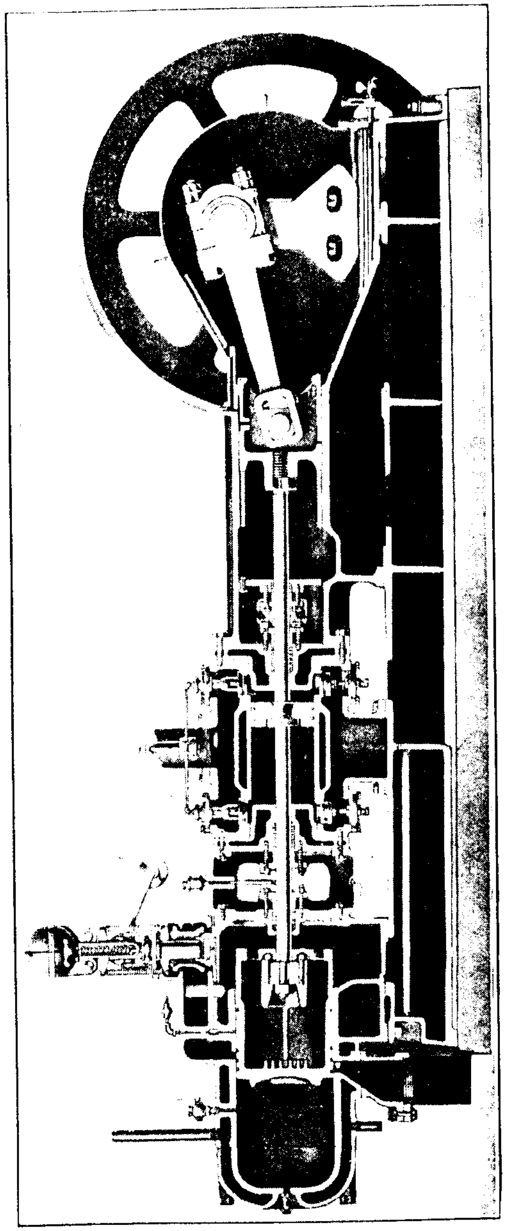

FIG. 5.—Chicago Pneumatic Tool Co. Straight-line, Fuel Oil-driven Compressor, Class N-SO2.

Holman Bros. build a Class P. I. of duplex, single-acting, high-speed compressors, for belt drive, in four sizes, with displacements of 82, 134, 212 and 272 cu.ft. per min., the first three sizes being also designed for electric drive.

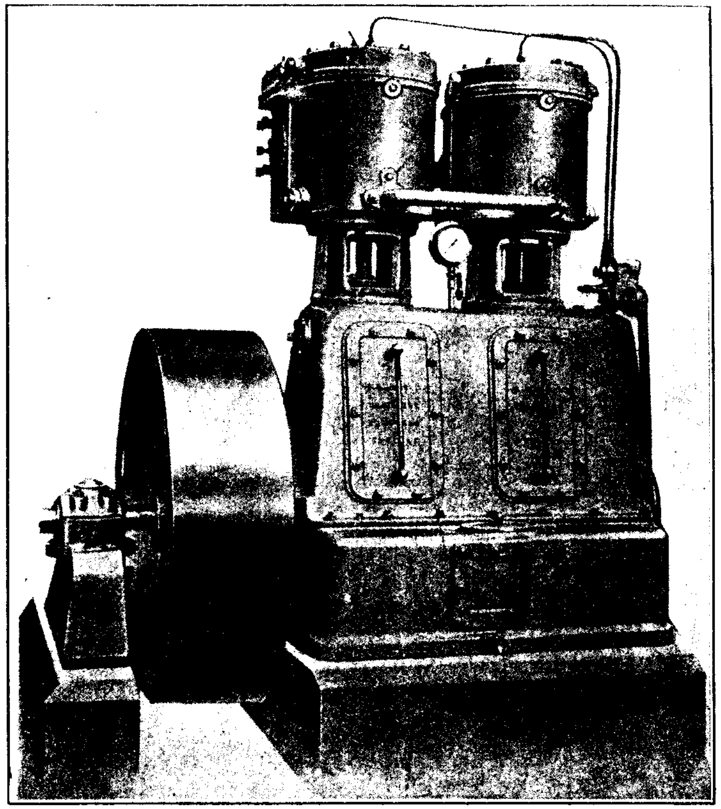

FIG. 6.—Holman Bros. Single-stage, Duplex, Double-acting Compressor.

The **Globe Pneumatic Engineering Co.** builds a line of 8 sizes of vertical, single-acting, direct-motor-driven compressors, with cylinders from 5 by 5 to 12 by 12 in. (Fig. 7). Somewhat similar compressors are made in four sizes by the Ingersoll-Rand Co., having cylinders from 3 by 3 to $4\frac{1}{2}$ by 5 in., with piston displacements of 7.3 to 23 cu.ft. per min. Small inclosed, single-acting, duplex compressors, direct-driven by electric motor or belt, are built also by the Allis-Chalmers Co. in five sizes. These are known as the Christensen compressor.

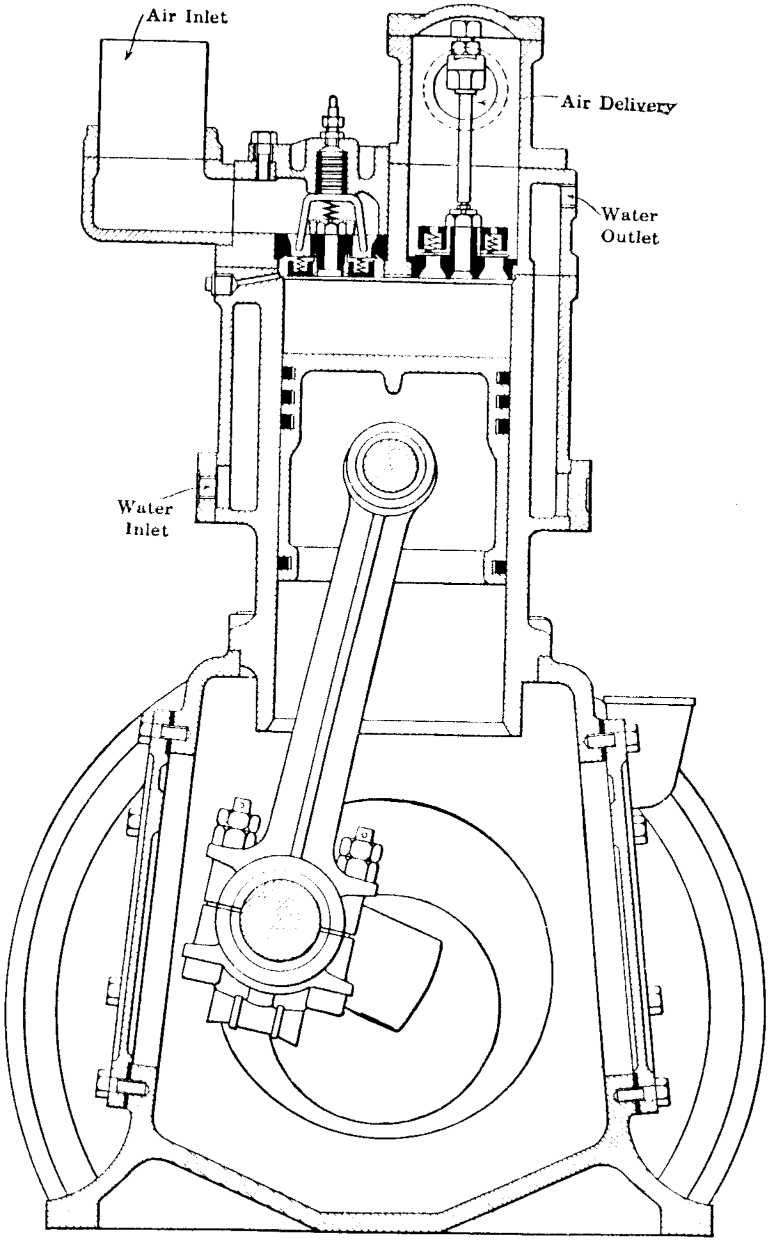

FIG. 7.—Globe Pneumatic Engineering Co. Single-acting, Vertical, Motor-driven Compressor.

Ingersoll-Rand Co. Straight-line, Steam-driven, Class FR-1 (Fig. 8)

Cylinder diam., in.			Stroke, in.	Rev. per min.	Displacement, cu.ft. per min.	Air press., lb.	I. Hp. steam cyl.	Floor space			
Steam		Air						Length,		Width,	
80–120 lb.	125–175 lb.							ft.	in.	ft.	in.
7	6	6	6	350	67	80–125	9–10	8	2	2	3
7	6	7	6	350	92	55–100	11–13	8	3	2	3
7	6	8	6	350	120	30–50	12–14	8	3	2	3
9	8	8	8	300	136	80–125	21–25	10	2	2	8
9	8	9	8	300	173	65–100	24–29	10	3	2	8
9	8	10	8	300	215	35–60	24–30	10	5	2	8
12	10	10	10	275	245	80–125	39–46	12	0	3	2
12	10	12	10	275	355	60–100	51–62	11	10	3	2
12	10	14	10	275	485	30–55	45–63	12	0	3	2
14	12	12	12	250	386	80–125	61–70	13	8	4	1
14	12	14	12	250	528	45–100	66–88	13	9	4	1

Note.—Seven other sizes of this class are made for air pressures of 10 to 40 lb.

Ingersoll-Rand Co. Straight-line, Belt-driven, Class ER-1 (Fig. 9)

Cylinders, in.		Rev. per min.	Displacement, cu.ft. per min.	Air pressure, lb.	
Diam.	Stroke			Min.	Max.
7	5	400	88	60	70
6	6	350	67	100	140
7	6	350	92	45	100
8	8	300	136	100	130
9	8	300	173	65	100
10	8	300	215	30	65
10	10	275	245	100	120
12	10	275	355	50	100
14	10	275	485	25	50
12	12	250	386	100	115
14	12	250	528	45	100

Note.—Smaller sizes of this class, having 4 by 4-in. to 7 by 5-in. cylinders, with displacements of 28 to 64 cu.ft. per min., are made; also, eight other sizes for air pressures of 10 to 45 lb.

All air cylinders of the FR-1 and ER-1 compressors are of the same design, the valves being set radially at the ends of the cylinder barrel. Small sizes have one inlet and one discharge valve at each end of the cylinder; the larger sizes, two or more inlet and discharge valves at each end, the number depending upon the volume of air compressed. Valves are readily removed for inspection by unscrewing four bolts in their bonnets. Normally, the air intake is at the bottom of the cylin-

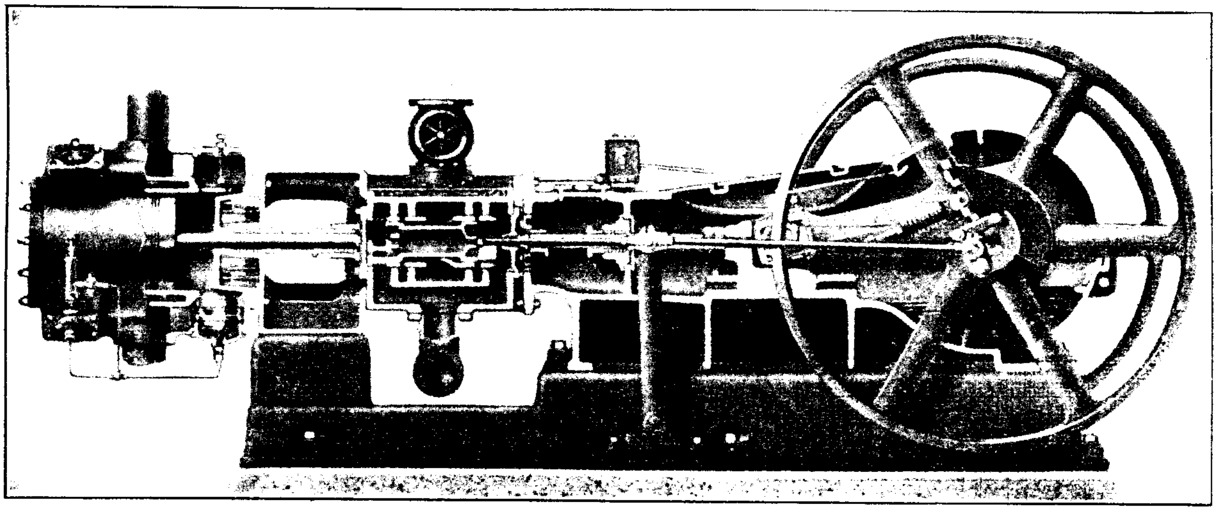

FIG. 8.—Ingersoll-Rand Co. Straight-line, Steam-driven Compressor. Class FR-1.

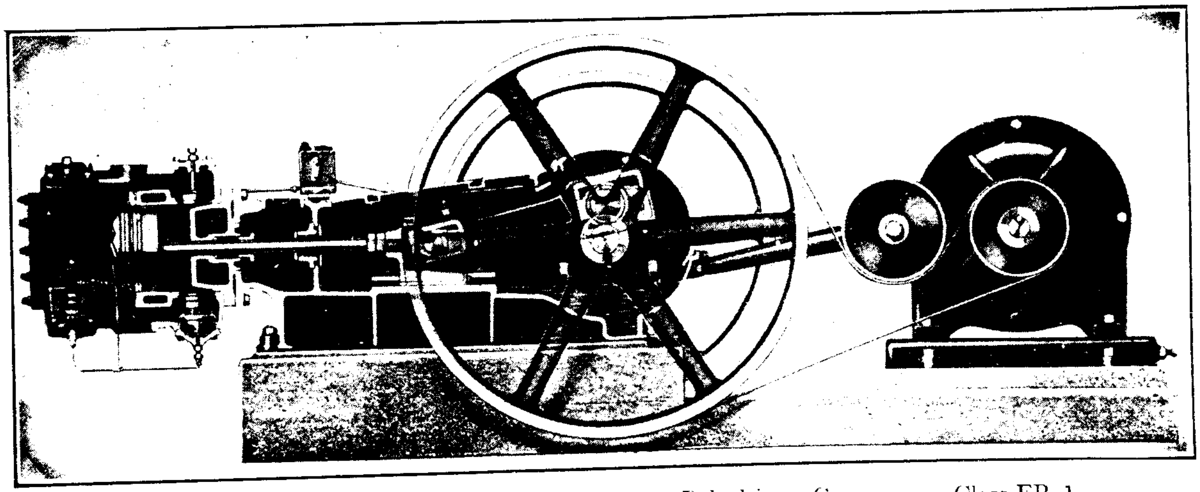

FIG. 9.—Ingersoll-Rand Co. Straight-line, Short Belt-driven Compressor, Class ER-1.

der, the discharge at the top. For compressors of 6-in. stroke and larger, this arrangement can, if desired, be reversed, by placing the inlet valves in the top half of the cylinder and the discharge valves in the lower half.

The Ingersoll-Rand Co. also builds two lines of single-stage, duplex compressors: XPV-1, with simple steam cylinders, and XPV-4, with cross-compound steam cylinders. For other forms of the XPV class, having two-stage air end, see Fig. 25 and accompanying text.

Norwalk Co. Duplex, Belt-driven, Type TB SD (Fig. 10)

Cylinders, in.		Rev. per min.	Displacement, cu.ft. per min.	Pressure, lb.	Brake hp.	Air pipe for both inlet and disch.	Floor space, in.	
Diam.	Stroke						Length	Width
6	6	350	134	125	28	2	74	54
7	6	350	184	100	33	2½	79	54
8	8	300	274	125	57	3	94	67
9	8	300	348	100	62	3	94	67
10	8	300	432	70	65	3½	95	67
10	10	275	492	100	88	4½	109	74¼
12	10	275	712	100	127	4½	109	74¼
14	10	275	969	60	131	5	111	74¼
12	12	250	774	100	138	5	124	83
14	12	250	1,060	100	189	5	126	83
16	12	250	1,390	60	188	6	126	83

Note.—The above compressors are driven by short belt. Five other sizes are built for pressures of 30 to 50 lb. per sq.in.; also, sixteen sizes of single-cylinder, belt-driven compressors (Type TB-S), with cylinders of same dimensions, the displacements being one-half the above.

The Norwalk Co. also builds the following: A duplex, single-stage, direct-motor-driven compressor (Type SB-SD), with 18 by 14-in. cylinder; displacement, 1,830 cu.ft. per min.; delivery pressure, 70 lb. Five sizes of the same type, with 20 by 14 to 28 by 16-in. cylinders; displacements, 2,265 to 4,520 cu.ft. per min.; delivery pressures, 50, 40 and 30 lb. Several sizes of straight-line, single-stage, steam-driven compressors (Type STB-S).

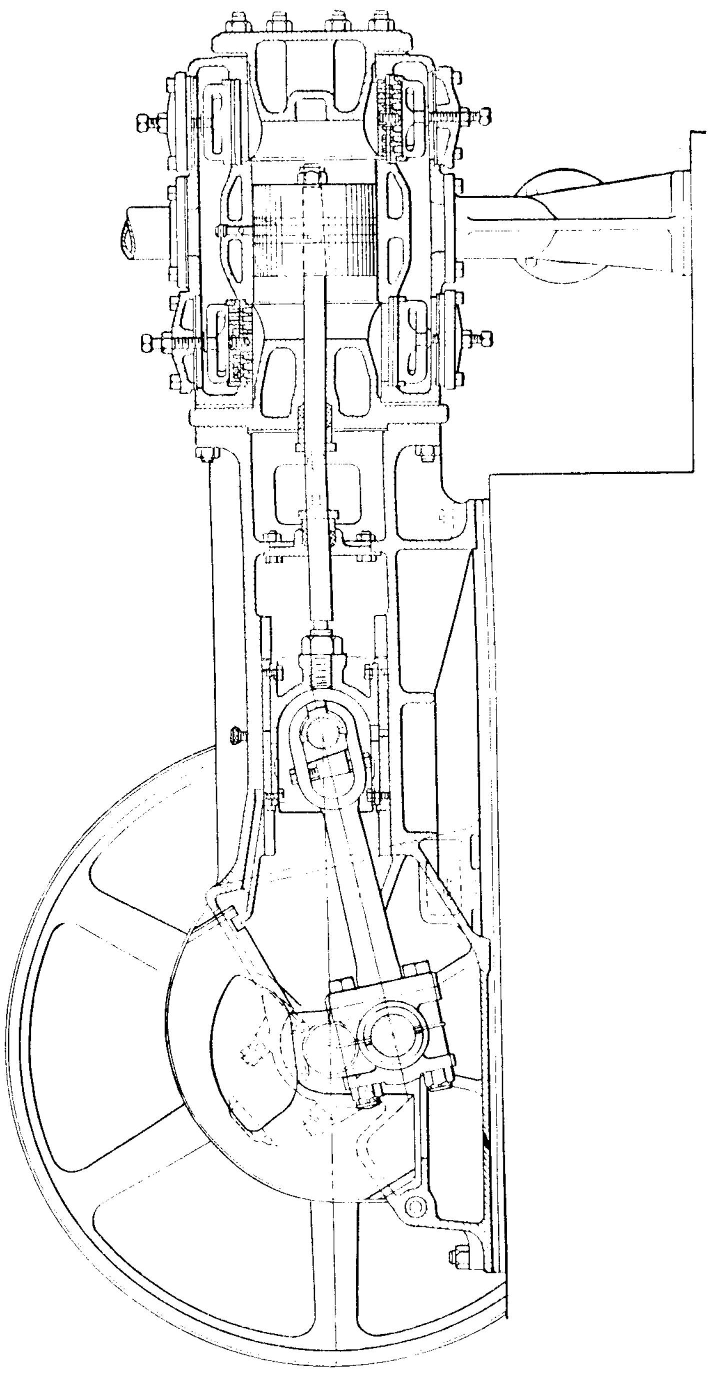

FIG. 10.—Norwalk Co. Duplex, Single-stage, Belt-driven Compressor, Type TB-SD.

Pennsylvania Pump & Compressor Co. Straight-line, Steam-driven, Class 4-A (Fig. 11)

Cylinder diam., in.			Stroke, in.	Rev. per min.	Displacement, cu.ft. per min.	Air pressure, lb.	I. Hp.	Floor space, in.	
Steam		Air						Length	Width
80 120 lb.	125 175 lb.								
7	6	6	6	350	67	75 125	9½-10	93	29
7	6	7	6	350	92	50 100	12 14	95	29
7	6	7½	6	300	106	40 90	12 15½	97	29
9	8	8	8	325	148	75 125	22 26	114	37
9	8	9	8	325	188	50 100	23-30	116	37
10	9	9	9	300	195	75 125	29 36	130	42
10	9	10	9	300	240	50 100	30-40	131	42
12	10	10	10	275	244	75 125	37-48	143	49
12	10	12	10	275	353	50 100	45-61	145	49
12	10	14	10	275	485	25 60	43 66	148	49
14	12	12	12	250	385	75 125	60 77	165	60
14	12	14	12	250	525	50 100	65 87	167	60
14	12	16	12	250	690	25 60	50 96	167	60

Note.—Twelve other sizes are built for pressures of 10 to 50 lb.

The Pennsylvania Pump & Compressor Co. also builds a line of belt-driven compressors, with displacements of 28 to 1,085 cu.ft. per min., those having the larger displacements being for low pressures (25 to 50 lb. per sq.in.).

Reavell & Co. Single-stage, Quadruplex, Belt-driven (Fig. 12)

For air pressures of 60 to 120 lb. per sq. in.

Size, in.	Max. rev. per min.	Max. displacement, cu.ft. per min.*	Suction opening, in.	Diam. delivery pipe, in.	†Brake hp. at compressor shaft for				Approx. floor space, direct-connected to electric motor, in.	
					60 lb.	80 lb.	100 lb.	120 lb.	Length	Width
6 by 4	600	153	4½ by 2¼	2	22	24	26	28	80	41
6½ by 4½	475	160	4½ by 2¼	2	24	26	28	31	81	42
7½ by 5	425	212	5½ by 2½	2½	29	32	35	38	83	43
8 by 6	400	274	5¾ by 2¼	3	38	42	46	50	87	45
9½ by 6	400	386	7 by 3¼	3½	55	63	69	73	105	51
11 by 7	350	530	7¼ by 3¾	4	75	87	96	103	116	53
12½ by 8	300	672	9 by 4½	4½	95	112	124	132	124	55
14½ by 9	275	932	10¼ by 5	5	132	155	173	184	137	64

* Free air actually delivered is 3 to 15 per cent less than piston displacement.

† Power required for lower pressures: 30 lb., 0.71 of that at 60 lb.; 50 lb., 0.9 of that at 60 lb. Delivery pressures of 70, 90 or 110 lb. may be had by regulating the speed of rotation.

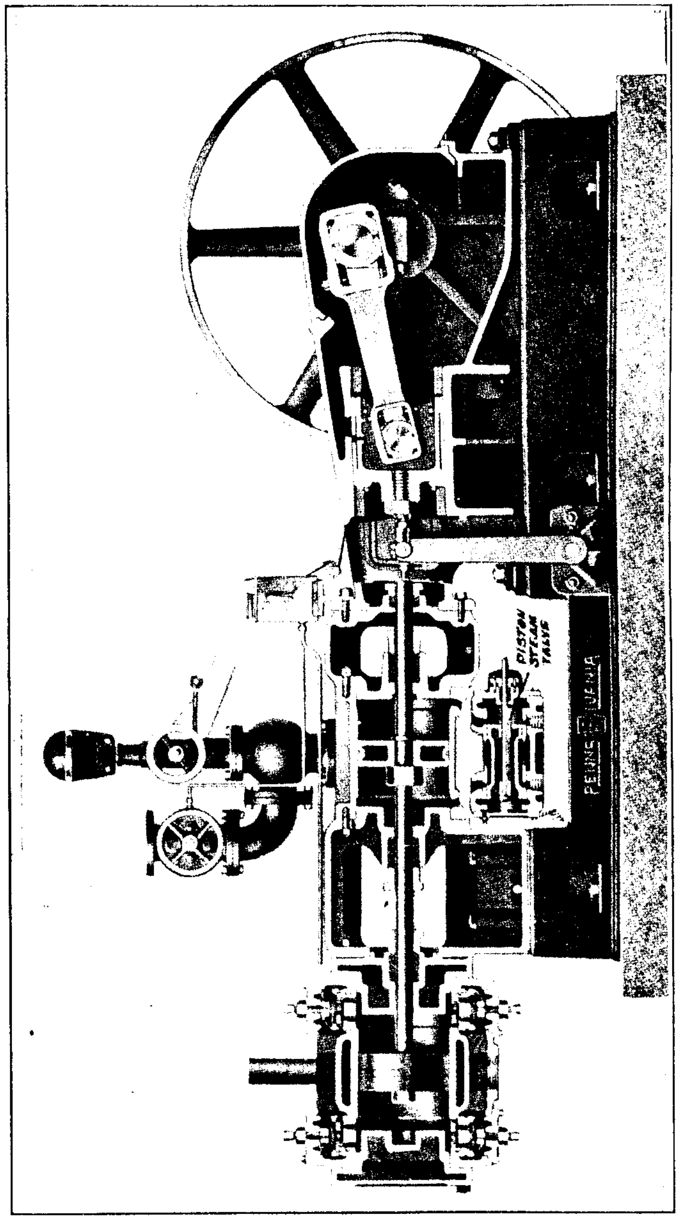

FIG. 11.—Pennsylvania Pump & Compressor Co. Straight-line, Steam-driven Compressor, Class 4-A.

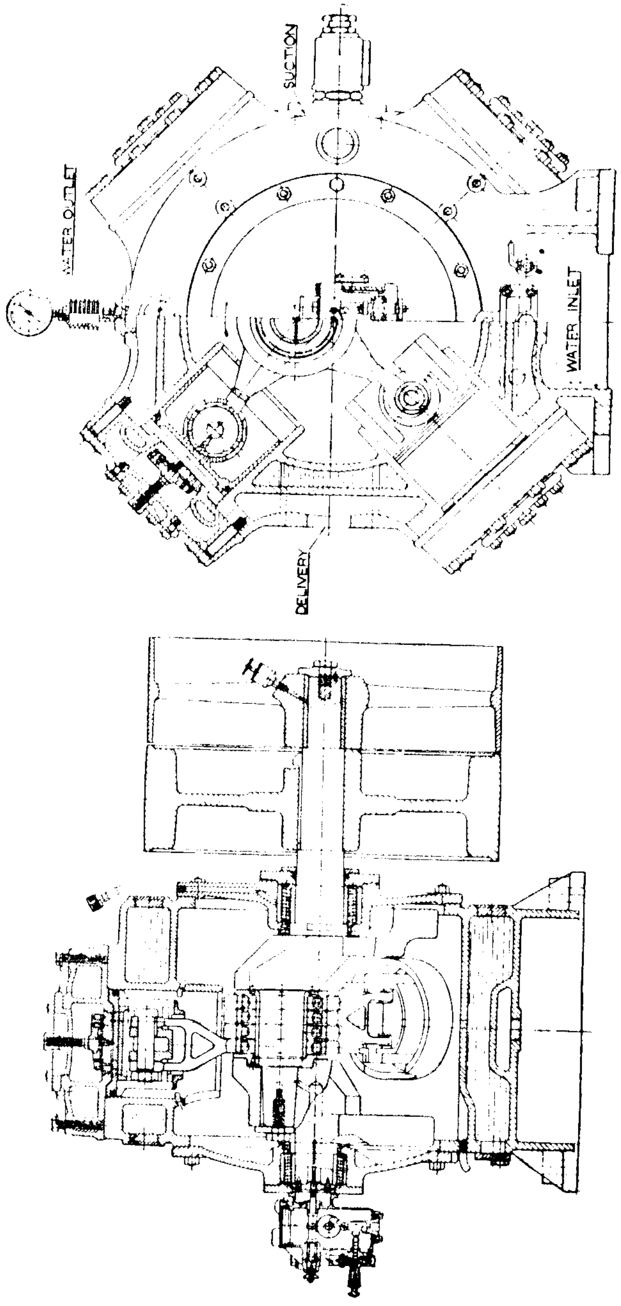

FIG. 12.—Reavell & Co. Single-stage, Quadruplex, Belt-driven Compressor.

The capacity of the Reavell Quadruplex compressor may be doubled by mounting a pair of units on a common bedplate, with a direct-connected electric motor, or a pulley for belt- or rope-drive between the units. An advantage of this peculiar design of compressor is that, even with a single unit, the variation in torque is small; and, in the case of the duplex form (pair of units), the torque becomes nearly constant.

Sullivan Machinery Co. Belt-driven, Class WG-6 (Fig. 13)

Cylinders, in.		Displacement, cu.ft. per min.	Rev. per min.	Air pressure, lb.	Hp.	Floor space				Air pipes, in.	
Diam.	Stroke					Length, ft.	in.	Width, ft.	in.	Inlet	Disch.
6	6	68	350	120	10.5	4	10½	1	10½	1½	1¼
7	6	93	350	100	12.5	5	0	1	10½	2	1½
8	6	122	350	50	14	5	5½	1	10½	2½	2
8	8	139	300	120	22.5	6	3	2	4½	2½	2
9	8	176	300	100	25	6	4	2	4½	3	2½
10	8	218	300	50	25.5	6	4	2	4½	3½	3
10	10	250	275	120	38.4	7	5	3	0	3½	3
11	10	302	275	100	48.3	7	6	3	0	4	3½
12	10	360	275	100	58.2	7	7	3	0	4	3½
14	10	490	275	50	60.8	7	7	3	0	5	4
12	12	377	240	120	68.5	8	10	3	11	4	3½
14	12	513	240	100	90.3	8	10	3	11	5	4
16	12	670	240	50	86.5	8	11	3	11	6	5

Note.—The Sullivan Company also makes similar straight-line, steam-driven compressors, Class WA-6 (Fig. 14). Five other sizes are built for air pressures of 15 to 30 lb.

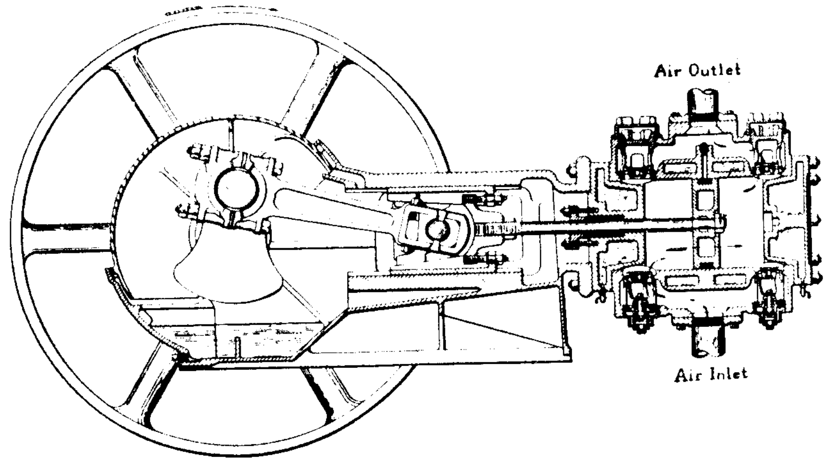

FIG. 13.—Sullivan Machinery Co. Single-stage, Belt-driven Compressor ("wafer" valves), Class WG-6.

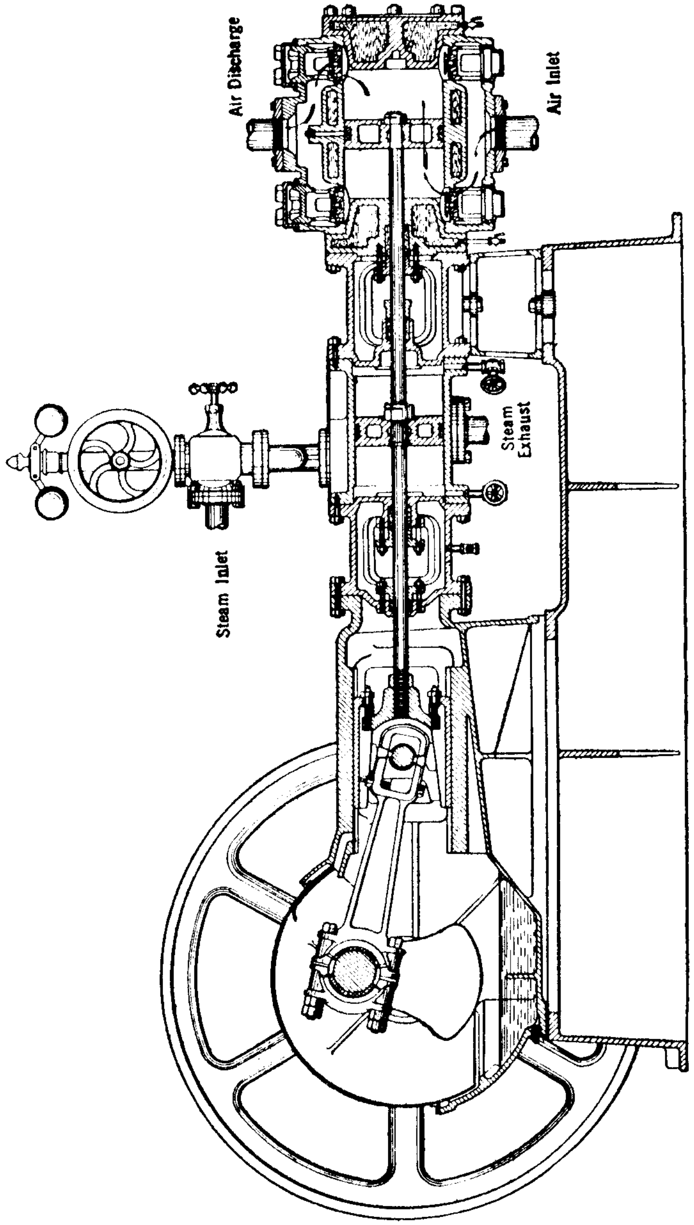

Fig. 14.—Sullivan Machinery Co. Straight-line, Single-stage, Steam-driven Compressor, Class WA-6.

Walker Bros. Horizontal, Single-stage, Duplex, Compound Condensing

Steam pressure, 120 lb.; air pressure, 60 lb.

Cylinder diam., in.			Stroke, in.	Rev. per min.	Capacity, cu.ft. free air per min.	Floor space	
Steam		Air				Length, ft. in.	Width ft. in.
H.P.	L.P.						
13	23	15	30	100	1,080	25 3	11 0
14	25	17	30	100	1,400	26 10	12 2
15	28	18	36	90	1,700	30 3	13 8
16	30	20	36	90	2,100	30 3	13 8
17	32	22	39	85	2,650	34 5	15 0
19	35	23	42	80	2,900	34 6	15 6
20	37	25	42	80	3,500	34 6	15 6
21	39	26	42	80	3,750	37 4	16 0
22	41	28	42	80	4,350	38 2	16 0
23	43	29	48	70	4,700	41 4	17 6
25	46	31	48	70	5,300	41 4	17 6
26	48	32	54	65	5,900	44 0	17 7
27	51	34	54	65	6,700	44 0	17 7
30	56	37	60	60	8,100	50 5	20 3
32	60	40	60	60	9,600	52 0	22 0

Note.—Simple, single-stage compressors, with same size air cylinders, and capacity as above, are also built, their steam ends being proportioned for pressures of 100, 120 and 140 lb.

Walker Bros. make a specialty of large horizontal two-stage compressors, of a variety of designs and sizes. An example is shown by Fig. 34.

Worthington Pump & Machinery Corp. Straight-line, Steam-driven (Fig. 15)

Cylinder diam., in.		Stroke, in.	Air pressure, lb.	Displacement, cu.ft. per min.	Rev. per min.	Floor space		Air pipes, in.	
Steam	Air					Length, ft. in.	Width, ft. in.	Inlet	Disch.
7	7½	6	80 100 125	106	350	7 0	2 3	2½	2½
8	8½	9	80 100-125	170	300	9 0	3 2	3	3
9	10	10	80 100-110	250	285	11 6	3 3	3½	3½
10	11	12	80 100-110	350	270	12 9	3 8	4	4
12	13	14	80-100	522	245	14 3	3 11	5	5
14	15	16	60-80	685	210	15 6	4 6	6	6
7	8½	6	40 60	136	350	7 2	2 3	3	3
8	10	9	40 60	245	300	9 3	2 9	3½	3½
9	11	10	40 60	300	275	11 9	3 6	4	4
10	13	12	40 60	475	260	13 0	3 8	5	5
12	16	14	40 60	782	245	14 6	3 11	6	6
14	18	16	40 60	981	210	15 9	4 6	8	8

Note.—The Worthington Company also makes a line of belt-driven compressors, of the same sizes as in the table. Six other sizes are built, for air pressures of 20 to 40 lb.

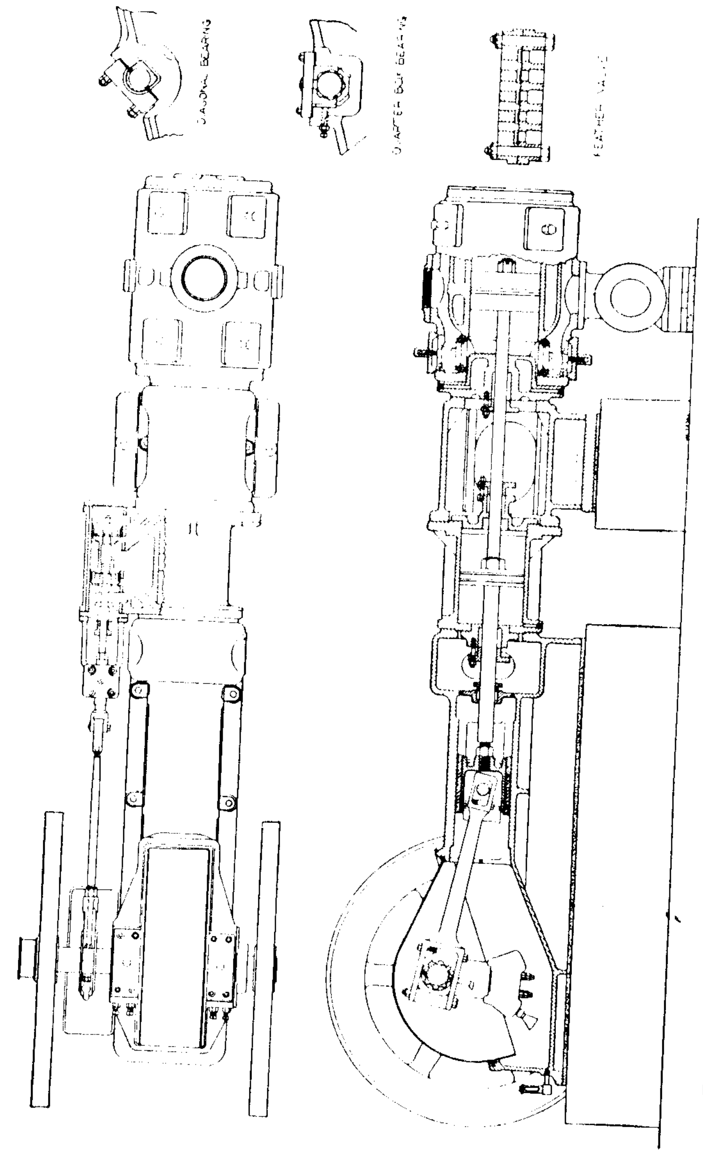

FIG. 15.—Worthington Pump & Machinery Corp. Straight-line, Steam-driven Compressor.

STAGE COMPRESSORS

These have two- to four-stage air cylinders, according to the pressure to be produced, the two-stage compressors being built in all of the structural forms included in the classifications near the beginning of this chapter.*

Two-stage compressors comprise larger sizes than single-stage machines, standard types having piston displacements to about 5,000 cu.ft. of free air per min., and producing pressures to say 150 lb. per sq.in. A few makers furnish units of still greater capacity. Two-stage compressors are now in common use for supplying air for rock-drills and other general service.

The three- and four-stage compressors are suitable for charging compressed-air locomotives for mine haulage; also for a variety of special service in the chemical industries, as the compression of oxygen, hydrogen, acetylene and other commercial gases. They are designed for pressures up to 5,000 lb. per sq. in. Data on the use of locomotive chargers are given in Chapter XXIV.

For ordinary pressures (to 125 or 150 lb.) the air cylinders of nearly all standard stage compressors are double-acting, *i.e.*, they have intake and delivery valves at each end of the cylinders. For high pressures, the cylinders of the higher stages are always single-acting, *i.e.*, they take air on one stroke and deliver on the following stroke. Details of the operation of stage compressors, with the modes of cooling the air during compression, are given in Chapter VI.

Several makers build large compressors only, more or less specialized in design; for example, the Allis-Chalmers Manufacturing Co. and the Nordberg Manufacturing Co. The former builds, to order, Corliss cross-compound, two-stage compressors, with displacements ranging from 3,000 to 10,000 cu.ft. of free air per minute (Fig. 16). One of these, with a piston displacement of 5,478 cu.ft., delivers 5,000 cu.ft. of free air per minute at 100 lb. pressure, when working at an altitude of 600 ft. (volumetric efficiency = 91.1 per cent). Steam cylinders, 18 and 40 in.; air cylinders, 22 and 38 in. diameter; stroke, 42 in.; initial steam pressure, 215 lb.; approximate weight, 30,000 lb.

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* It is of historical interest to note that the Norwalk Iron Works Co., now the Norwalk Co., was the pioneer in the field of stage compression, having begun in 1880–81 to build this type of compressor for ordinary service.

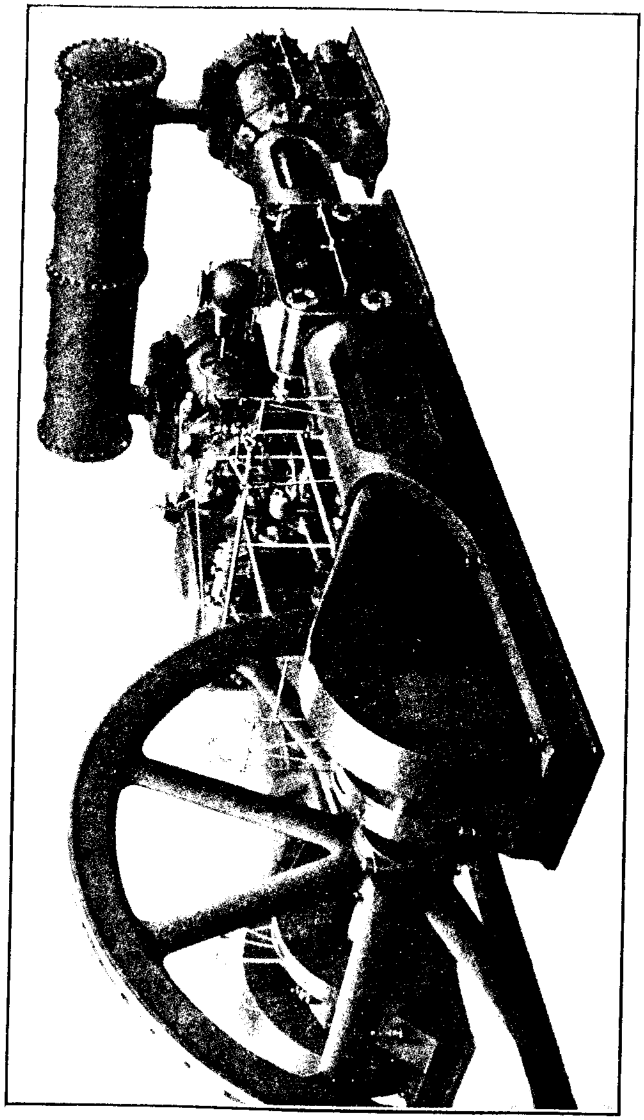

Fig. 16.—Allis-Chalmers Manufacturing Co. Cross-compound, Two-stage Compressor.

W. H. Bailey & Co. Horizontal, Differential Two-stage, Belt-driven, Class L

Differential-piston diam., in.		Stroke, in.	Rev. per min.	Actual capacity per min., cu.ft.	Brake hp., max.		Air pipes, in.		Floor space, in.	
L.P.	H.P.				For 85 lb.	For 100 lb.	Inlet	Disch.	Length	Width
$10\frac{1}{4}$	$8\frac{5}{8}$	$7\frac{7}{8}$	225	78	11.5	16	3.5	2	63	49
$11\frac{13}{16}$	$10\frac{1}{8}$	$9\frac{7}{8}$	210	126	23	25	4	2.5	73	54
$13\frac{3}{8}$	$11\frac{3}{8}$	$11\frac{13}{16}$	200	178	33	36	4.5	2.5	88	63
$15\frac{3}{4}$	$13\frac{3}{8}$	$13\frac{3}{4}$	180	260	49	51	6	3	96	64
$17\frac{3}{4}$	$15\frac{1}{8}$	$15\frac{1}{4}$	160	335	61	65	7	3.5	105	73
$19\frac{11}{16}$	$15\frac{1}{8}$	$15\frac{3}{4}$	160	415	75	80	9	5	105	73
$21\frac{7}{8}$	$17\frac{3}{4}$	$17\frac{3}{4}$	150	530	96	105	10	6	118	87
$24\frac{1}{4}$	$18\frac{5}{16}$	$17\frac{3}{4}$	167	735	125	140	11	7	122	95
27	$20\frac{1}{4}$	$19\frac{11}{16}$	150	885	160	175	12	8	130	102

Note.—All the above sizes are suitable for air pressures of 70 to 150 lb. per sq. in.

In the Bailey compressors specified in the table two-stage compression is effected in a single cylinder by the use of a differential piston, with no more parts than are required for a single-stage compressor. Fig. 17 shows an unusual design by the same makers. It is a vertical, two-stage compressor, with a differential air piston; capacity, 950 cu.ft. free air per min., at 155 rev. per min. They also build fourteen sizes of vertical, two-stage, belt-driven compressors, delivering 159 to 2,000 cu.ft. free air per min. at 85 to 100 lb. pressure, the brake horsepowers ranging from 28 to 382.

Chicago Pneumatic Tool Co. Cross-compound, Steam-driven, Class O-CSC (Fig. 18)

Cylinder diam., in.				Stroke, in.	Rev. per min.	Piston displacement, cu.ft. per min.	Max. air pressure, lb.	I. hp. of steam cylinders at air press. shown		
Steam		Air								
H.P.	L.P.	Intake	Disch.					80 lb.	100 lb.	125 lb.
10	16	15	9	10	250	511	150	86	96	106
12	19	17	10	12	235	741	125	120	135	147
14	22	20	12	14	225	1,145	125	175	194	213
16	26	23	13	16	225	1,730	125	231	254	281
16	26	24	14	18	200	1,884	125	262	288	318
16	26	26	15	18	200	2,212	125	360	394	437

Note.—Class O-DSC two-stage compressors have the same size air cylinders as above, and duplex single-expansion steam cylinders from 9 to 16 in. diam.

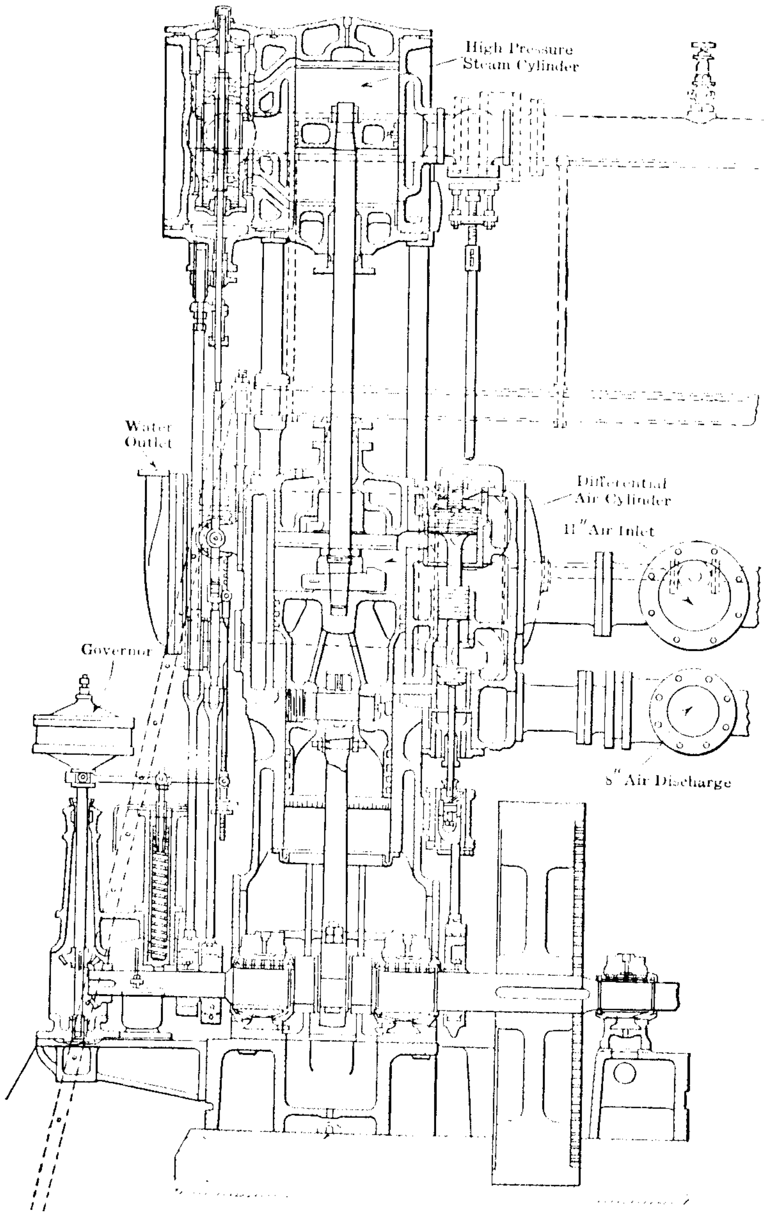

Fig. 17.—W. H. Bailey & Co. Vertical, Duplex, Cross-compound, Two-stage Compressor, with Differential Air Piston. Vertical Section through High-pressure Steam Cylinder.

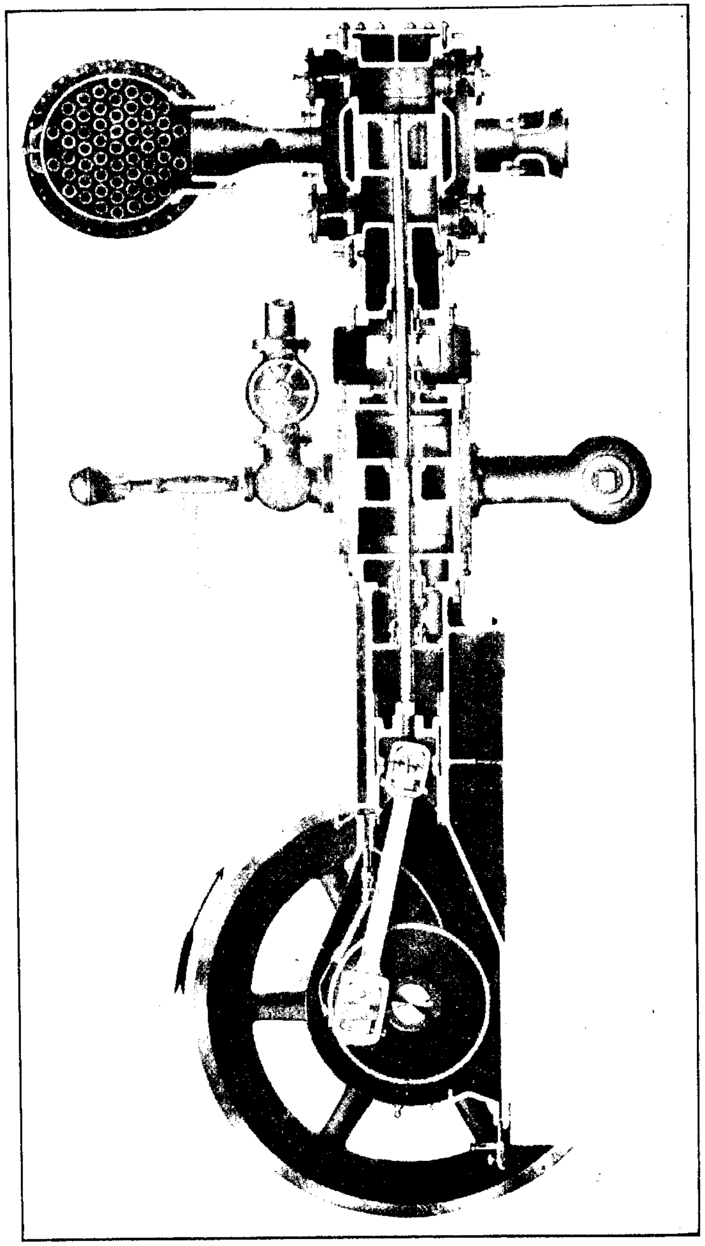

Fig. 18.—Chicago Pneumatic Tool Co. Two-stage, Cross-compound Compressor, Class O–CSC. (Longit. section through high-pressure steam cylinder.)

Chicago Pneumatic Tool Co. Duplex, Two-stage, Direct-connected to Synchronous Motor, Class O-CE

Cylinder diam., in.		Stroke, in.	Piston displacement, cu.ft. per min.	Rev. per min.	Max. pressure, lb. per sq. in.	Air pipes, in.		Approx. floor space	
L.P.	H.P.					Inlet	Disch.	Length, ft. in.	Width, ft. in.
15	9	12	675	277	125	5	5	9 8	9 2
17	10	12	806	257	125	6	6	10 6	10 3
19	11	12	1,007	257	110	8	6	10 6	10 6
20	12	14	1,300	257	125	8	8	12 0	11 9
23	13	16	1,722	225	125	10	10	13 8	11 8
26	15	18	2,200	200	125	12	12	16 8	14 8
30	18	21	3,074	180	125	12	12	17 2	15 0

The Chicago Pneumatic Tool Co. also builds the following: (*a*) Class O-CB, six sizes of duplex, two-stage, belt-driven compressors, with cylinders from 15 and 9 in. by 10-in. stroke to 23 and 13 in. by 16 in.; displacements, 609 to 1,722 cu.ft. free air per min.; air pressures, 110 to 125 lb. (*b*) Class O-CTDE (Fig. 19), three sizes of tandem duplex, two-stage compressors, driven by direct-mounted synchronous motor; cylinders, 20 and 12 by 16-in. stroke to 26 and 15 by 21-in. stroke; displacements, 2,577 to 4,596 cu.ft. per min.; maximum air pressure, 125 lb. (*c*) A line of duplex, belt-driven compressors, with delivery pressures of 30 to 60 lb.; cylinders, 15 by 10 in. to 24 by 18 in.; displacements, 1,022 to 3,768 cu.ft. per min.

Dujardin et Cie. Horizontal, Duplex, Two-stage, Type CHD (rope-driven)

Delivery pressure, 100 lb.

Cylinder diam., in.*		Stroke, in.	Rev. per min.	Piston displacement	Actual air intake	Hp.	Approx. floor space, ft.†	
L.P.	H.P.			Cu.ft. per min.			Length	Width
$19\frac{1}{4}$	$11\frac{3}{4}$	$19\frac{5}{8}$	211	1,110	1,134	224	17	11
24	$14\frac{9}{16}$	$19\frac{5}{8}$	187	1,903	1,593	310	18	12
$26\frac{3}{8}$	$15\frac{11}{16}$	$21\frac{5}{8}$	187	2,538	2,119	415	20	14
$30\frac{5}{16}$	$18\frac{1}{16}$	$21\frac{5}{8}$	187	3,388	2,821	553	21	16
$33\frac{7}{16}$	$20\frac{1}{16}$	$27\frac{1}{2}$	150	4,239	3,537	692	22	18
$36\frac{7}{16}$	22	$27\frac{1}{2}$	150	5,049	4,239	830	23	19
$40\frac{1}{2}$	24	$33\frac{7}{16}$	125	6,264	5,292	1,038	25	22
$43\frac{1}{4}$	$26\frac{3}{8}$	$43\frac{1}{4}$	100	7,425	6,345	1,245	28	24

* The cylinder dimensions are converted from millimeters to inches.
† These dimensions include space occupied by the rope pulley.

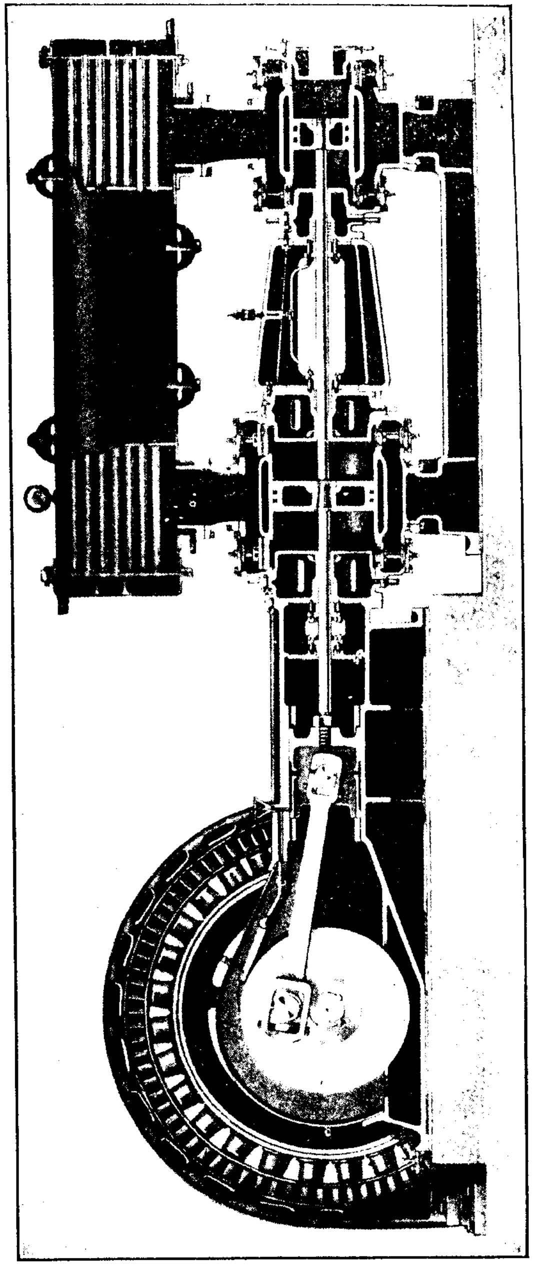

FIG. 19.—Chicago Pneumatic Tool Co. Two-stage, Tandem Duplex, Motor-driven Compressor, Class O-CTDE.

The above compressors are also built with cross-compound steam end (Fig. 20), and for electric drive. Dujardin & Cie also build the following lines of compressors: horizontal, two-stage, motor-driven, Type CH; vertical, duplex, two-stage, with differential pistons; horizontal, five-stage.

Etablissements François. Horizontal, Duplex, Two-stage, Belt-driven, Type BC (Fig. 21)

Cylinder diam.				Stroke		Rev. per min.	Piston displacement		Approx. hp. for 90 lb. pressure
L.P.		H.P.							
Mm.	In.	Mm.	In.	Mm.	In.		Cu. meters per hr.	Cu. ft. per min.	
310	12.20	185	7.28	300	11.80	250	680	400	70
375	14.56	225	8.85	300	11.80	250	990	582	100
425	16.73	255	10.04	350	13.77	250	1,150	869	150
525	20.66	305	12.00	400	15.74	214	2,200	1,292	230
560	22.04	335	13.18	500	19.68	214	3,150	1,854	320
570	22.43	340	13.38	400	15.74	214	2,600	1,530	265
625	24.60	365	14.36	500	19.68	214	3,950	2,325	400
695	25.36	415	16.33	500	19.68	214	4,850	2,853	500
830	32.67	500	19.68	600	23.61	166.6	6,450	3,796	650

Note. The smaller sizes are also built in straight-line (tandem) form.

Holman Bros. Vertical Compound, Two-stage, Type B.Y.2 (Fig. 22)

Cylinder diam., in.				Stroke, in.	Rev. per min.	Displacement, cu.ft. per min.	Air pipes, in.		Floor space, in.	
Steam		Air								
H.P.	L.P.	L.P.	H.P.				Inlet	Disch.	Length	Width
8½	14	12	7½	7	380	348	4	2½	93	69
12	20	16½	10	8	340	670	5	3½	120	87
14½	22	19¾	12	9	320	1,012	7	4	138	98
16	24½	23	14	10	290	1,380	8	4½	156	109

Approximate weights 15,700, 31,300, 50,000 and 68,800 lb. respectively. This firm also builds Type B.Y.2 compressors for belt, rope or electric drive.

Lines of two-stage vertical compressors, with both steam and direct electric drive, are also built by the following British makers: Belliss & Morcom (Fig. 23), Reavell & Co. (Fig. 24), Robey & Co. and Walker Bros. (see Fig. 35).

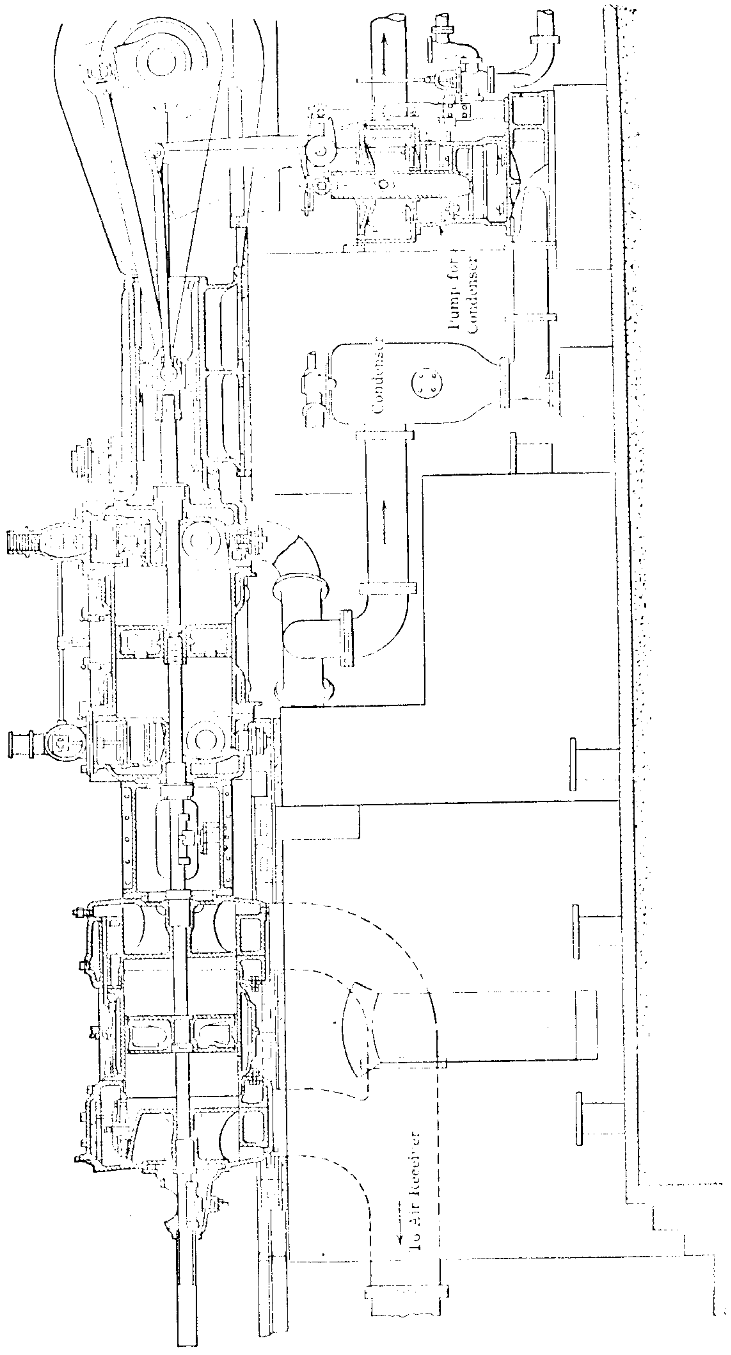

FIG. 20.—Dujardin et Cie. Cross-compound, Two-stage Compressor (section through low-pressure cylinder). Capacity, 4,120 cu.ft. free air per min.

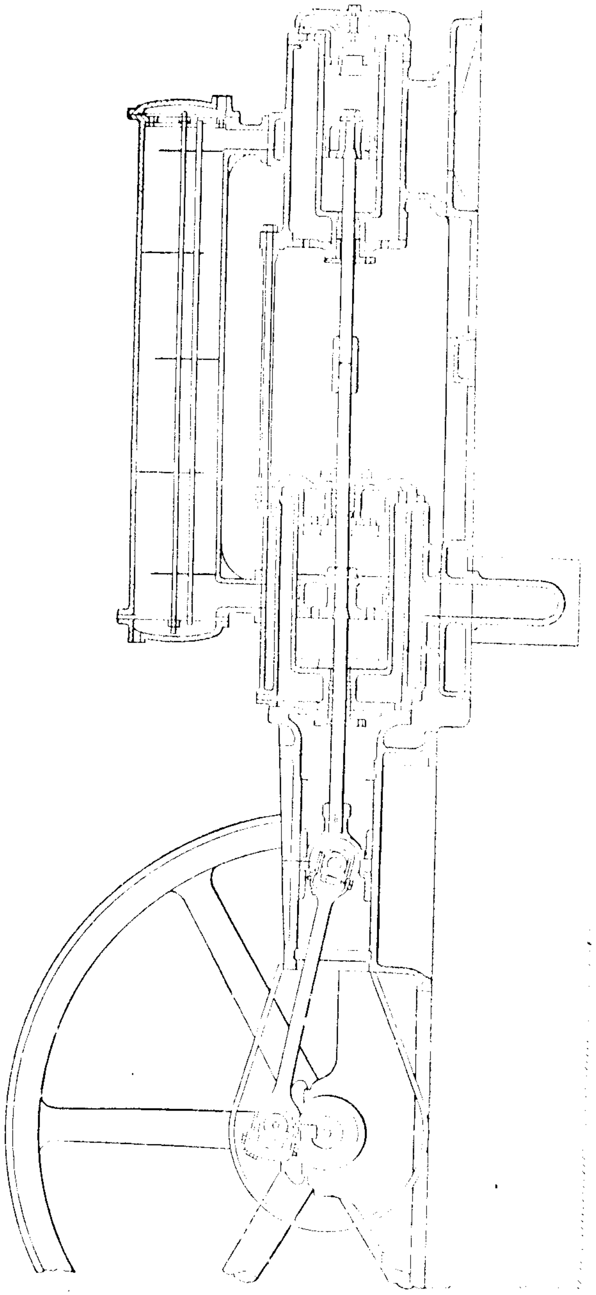

Fig. 21.—Etablissements Français. Horizontal, Two-stage, Belt-driven Compressor, Type BC.

The Ingersoll-Rand Class XPV compressors are built in four forms: XPV-1, duplex, with single-stage air cylinders and simple steam end; XPV-2, duplex, with two-stage air cylinders and simple steam end; XPV-3 (Fig. 25), with two-stage air and cross-compound steam end;

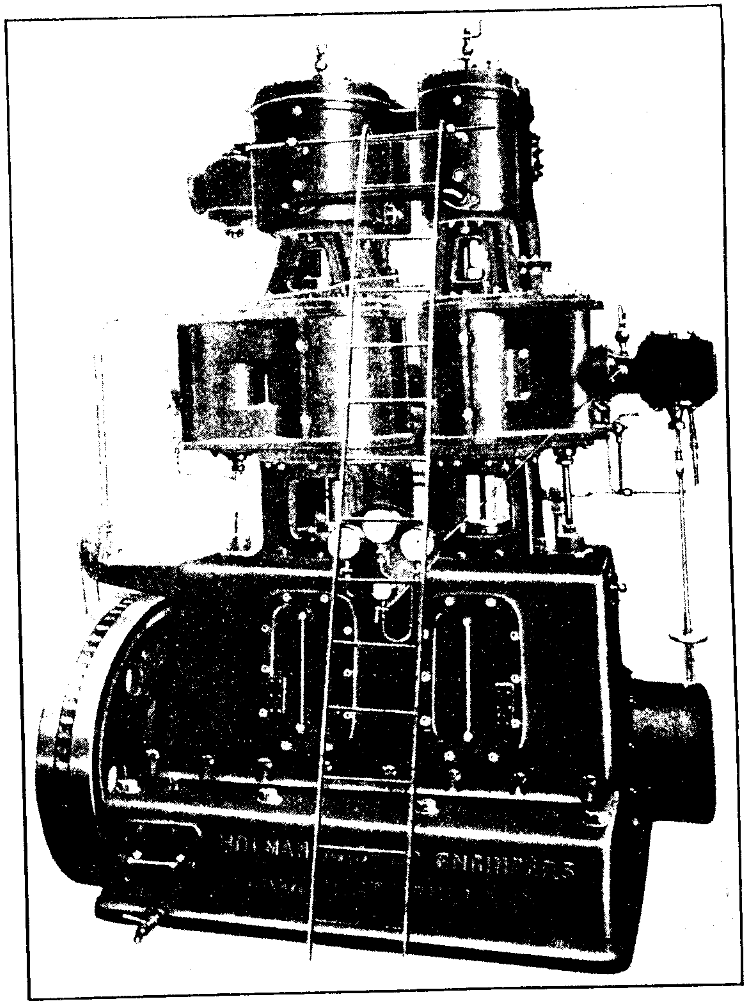

Fig. 22.—Holman Bros. Vertical Compound, Two-stage Compressor, Type B.Y. 2.

XPV-4, with single-stage duplex air and cross-compound steam end. Compressors of these general types are also built for belt and direct-connected motor drive. The accompanying table of specifications of the XPV-2 class shows the different proportions of the air cylinders for operation at sea level and at altitudes of 5,000 and 10,000 ft. Formerly,

Ingersoll-Rand Co. "Imperial" XPV-2, Duplex Steam and Two-stage Air Cylinders

Air pressure, 15-100 lb.; steam pressure, 80-120 lb.

For sea-level				For 5,000-ft. altitude				For 10,000-ft. altitude				For all altitudes	
Cylinder diam., in.			Piston displacement, cu.ft. per min.	Cylinder diam., in.			Piston displacement, cu.ft. per min.	Cylinder diam., in.			Piston displacement, cu.ft. per min.	Stroke, in.	Rev. per min.
Steam	L.P. air	H.P. air		Steam	L.P. air	H.P. air		Steam	L.P. air	H.P. air			
7	11	7	246	7	12	7	293	7	13	7	344	10	225
8	13	8	412	8	14	8	479	8	15	8	550	12	225
9	15	9¼	549	9	16	9¼	625	9	17	9¼	706	12	225
10	17	10½	823	18	18	10½	923	10	19	10½	1,029	14	225
12	20	12½	1,156	12	21	12½	1,275	12	22	12½	1,406	16	200
14	22	14	1,400	14	23	14	1,551	14	24	14	1,667	16	200
15	23	14	1,723	15	24	14	1,877	15	25	14	2,037	20	180
16	26	16	2,200	16	27	16	2,373	16	28	16	2,553	20	180
19	30	19	3,026	19	32	19	3.444	19	34	19	3,890	24	155
22	35	22	4,150	22	37	22	4,640	22	39	22	5,157	30	125

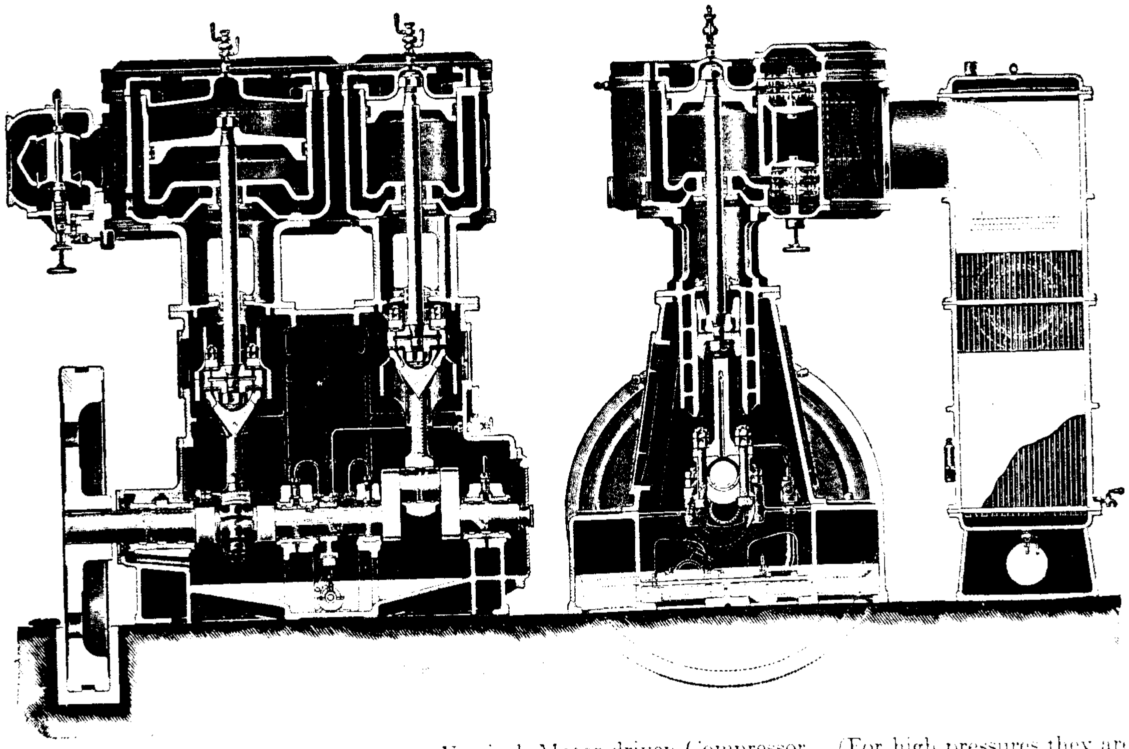

FIG. 23.—Bellis & Morcom. Two-stage, Vertical, Motor-driven Compressor. (For high pressures they are 3- or 4-stage.)

the air cylinders of these compressors had Corliss inlet and poppet discharge vales, and many are still in use. They are now fitted with

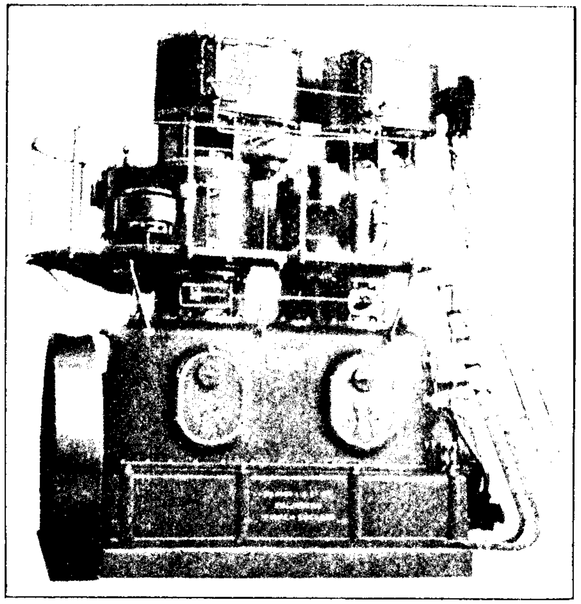

Fig. 24.—Reavell & Co. Two-stage, Vertical, Steam-driven Compressor. (Made in eleven sizes, delivering 500–5,000 cu.ft. free air per min. at 120 lb.)

"plate" valves, for both inlet and discharge, for details of which, see Chapter VII.

Ingersoll-Rand Co. "Imperial" Two-stage, Duplex, Belt-driven, Class XCB 2

Cylinders proportioned for different altitudes; air pressure, 80–110 lb.

For sea-level			For all altitudes		For 5,000-ft. altitude			For 10,000-ft. altitude		
Cylinder diam., in.		Displacement, cu.ft. per min.	Stroke, in.	Rev. per min.	Cylinder diam., in.		Displacement, cu.ft. per min.	Cylinder diam., in.		Displacement, cu.ft. per min.
L.P.	H.P.				L.P.	H.P.		L.P.	H.P.	
15	9¼	676	12	277	16	9¼	770	17	9¼	870
17	10½	807	12	257	18	10½	905	19	10½	1,008
18	11	1,052	14	257	19	11	1,173	20	11	1,306
20	12½	1,302	16	225	21	12½	1,435	22	12½	1,575
22	14	1,575	16	225	23	14	1,723	24	14	1,875

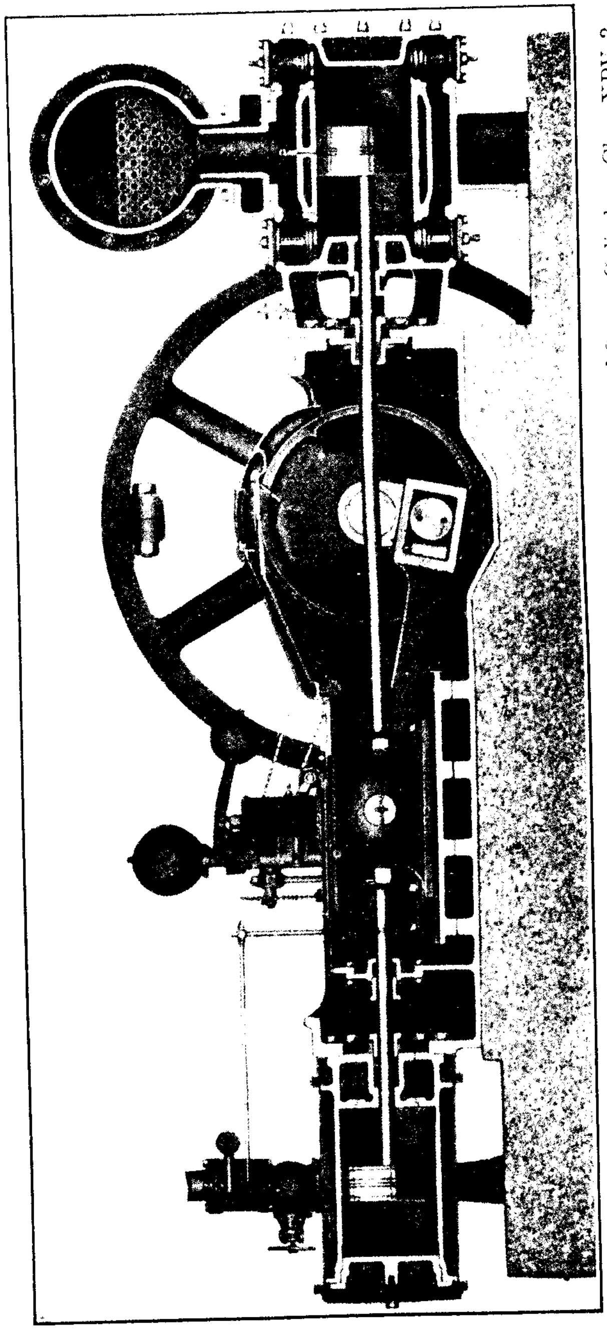

Fig. 25.—Ingersoll-Rand Co. Two-stage, Duplex Compressor, with Cross-compound Steam Cylinders, Class XPV-3.

The Ingersoll-Rand two-stage compressors of Class XRE-2 have the same cylinder dimensions as those specified in the table of "Imperial" compressors, but are direct-connected to synchronous motors. Those having low-pressure cylinders of 15, 17 and 18 in. diameter are designed for service at sea-level; those of 16, 18 and 19 in., for 5,000-ft. altitude. Class PRE-2 of the same maker (Fig. 26), also direct-connected to synchronous motors, are built in fifteen sizes, from 20 by 12½ by 14 in. to 46 by 28 by 36 in.; their piston displacements (cu.ft. free air per min., approximately) are from 1,302 to 7,140 at sea-level and 1,435 to 8,145 at 5,000 ft. altitude. All compressors of Classes XCB, XRE and PRE are fitted with five-step clearance control (see Chap. VIII).

Ingersoll-Rand Co. Two-stage, Fuel Oil-driven, with Differential, Single-acting Air Cylinder, Class POC 2 (Fig. 27)

Unit type*	Displacement, cu.ft. per min.	Delivery pressure, lb. per sq.in.†		Floor space				Height, ft. in.		Approx. shipping wt., lb.
		Normal	Max.	Length, ft.	in.	Width, ft.	in.	ft.	in.	
POV 2	354	100	125	10	6	4	4	9	4	15,750
POC 2	603	100	125	18	2	6	0	7	6½	35,000
POC 2	435	300	500	18	0	6	0	6	9	31,800
POC 2	895	100	125	22	6	8	3	7	11	58,000

* In Class POV, the air cylinder is set vertically (Fig. 28); in Class POC, it is horizontal.
† Single-stage compressors, for lower air pressures, are also made.

Norwalk Co. Straight-line, Two-stage, Belt-driven, Type TB-S2T (Fig. 29)

For 110 lb. air pressure

Cylinder diam., in.			Rev. per min.	Displacement, cu.ft. per min	Brake hp.	Air pipes, in.		Floor space, in.	
L.P.	H.P.	Stroke				Inlet	Disch.	Length	Width
10	6	8	300	219	40	4	2.5	133	37
12	7.5	8	300	315	58	4.5	2.5	135	37
12	7.5	10	275	355	67	4.5	2.5	156	44
14	8	10	275	490	90	5	3.0	156	44
15	9	12	250	600	114	6	4.5	175	52
16	10	12	250	700	129	6	4.5	175	52
18	10	12	250	885	164	7	4.5	177	52

The Norwalk Co.'s type SB-S2X, built in duplex form, with displacements of 920 to 2,280 cu.ft. per min., are driven by direct-connected synchronous motors. Types TB-S2TSA (belt-driven) and STB-S2TSA

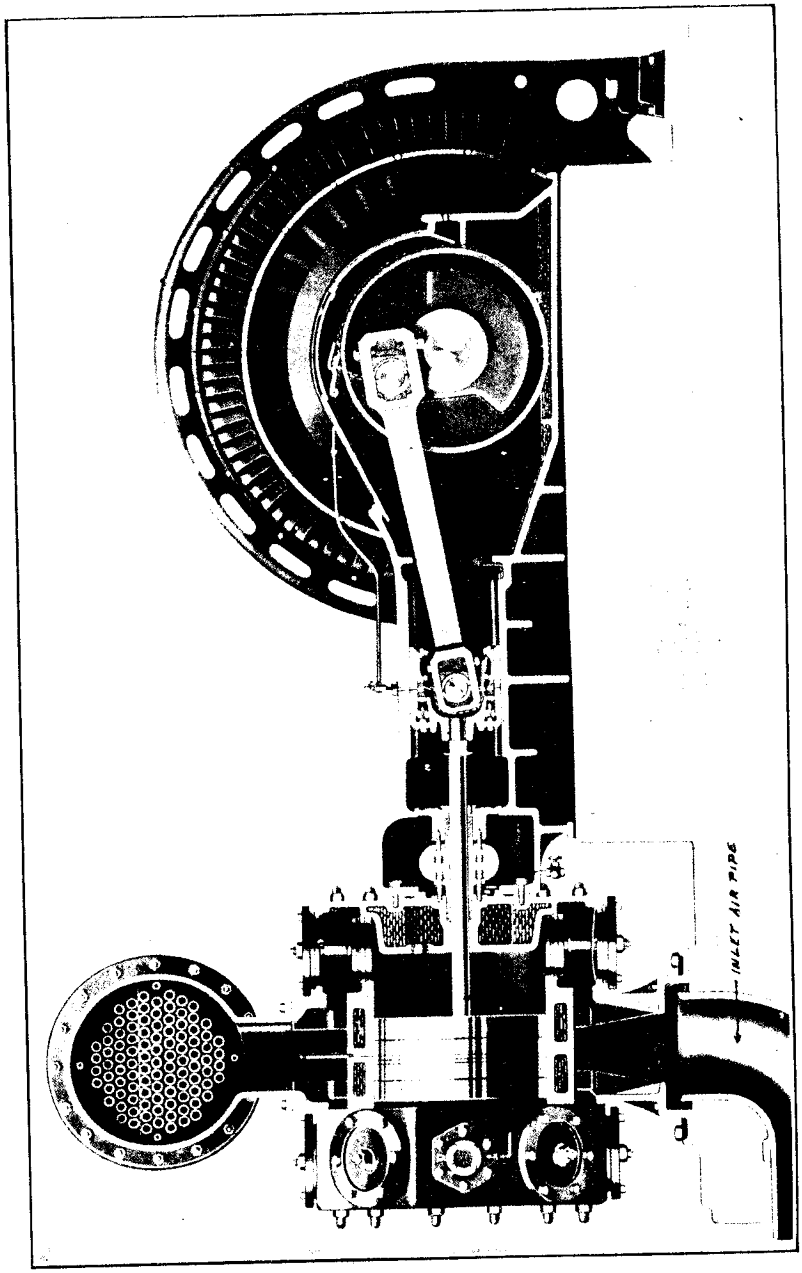

FIG. 26.—Ingersoll-Rand Co. Duplex, Two-stage, Electric-driven Compressor, Class PRE-2.

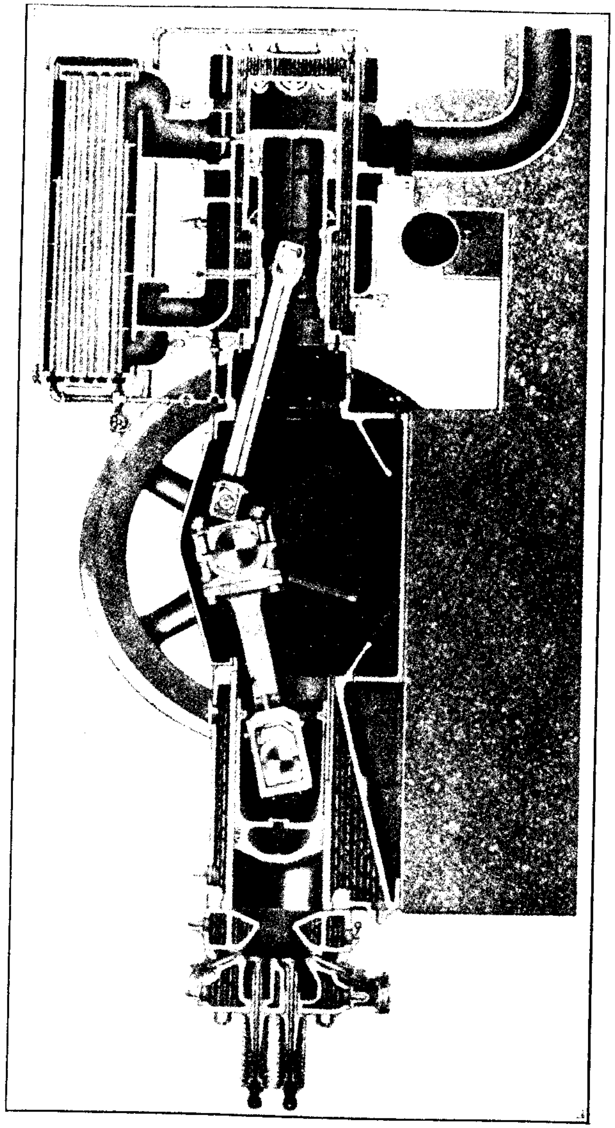

FIG. 27.—Ingersoll-Rand Co. Two-stage, Fuel Oil-driven Compressor, with Horizontal, Single-acting, Differential Air Cylinder, Class POC-2.

(steam-driven), designed for pressures of 200 to 500 lb., are built in fourteen sizes, with intake cylinders of 6 by 6 in. to 14 by 12 in., and displacements of 23 to 314 cu.ft. per min. The two largest sizes are duplex.

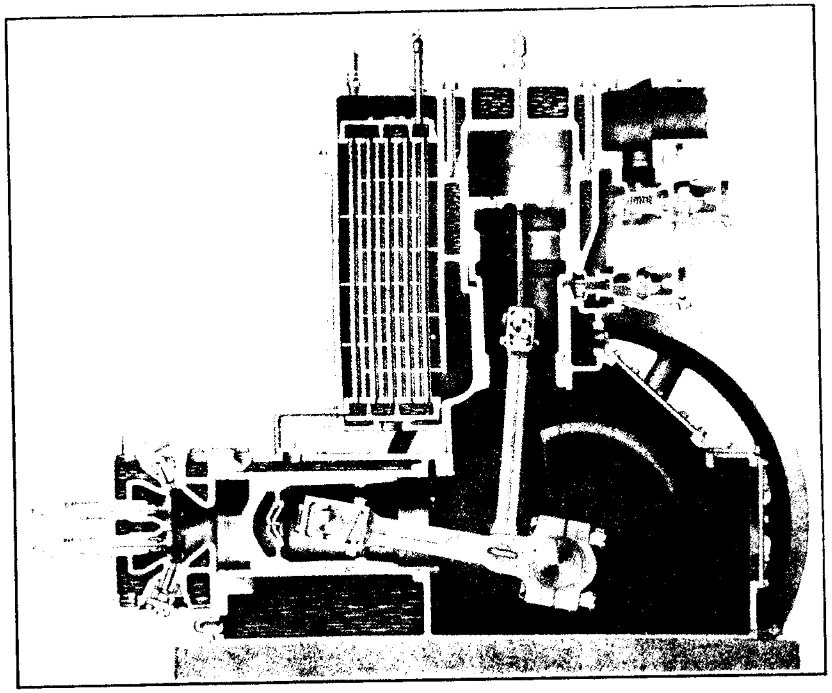

FIG. 28.—Ingersoll-Rand Co. Two-stage, Fuel Oil-driven Compressor, with Vertical, Differential Single-acting Air Cylinder, Class POV-2.

Norwalk Co. Belt-driven, Three-stage, Type TB-S3T (Fig. 30)

Suitable for charging compressed-air locomotives

Intake cylinder, in.		Displacement, cu.ft. per min.	Brake hp.				Floor space, in.	
Diam.	Stroke		For 500 lb.	For 1,000 lb.	For 1,500 lb.	For 2,000 lb.	Length	Width
7	6	32	9.0	11.5	13.0	14.2	100	29
9	8	70	19.8	25.2	28.4	31.0	124	37
10	10	112	31.6	40.5	45.5	49.5	144	44
12	10	158	44.5	57.0	64.3	70.0	144	44
12	12	190	53.6	68.5	77.0	83.0	154	52

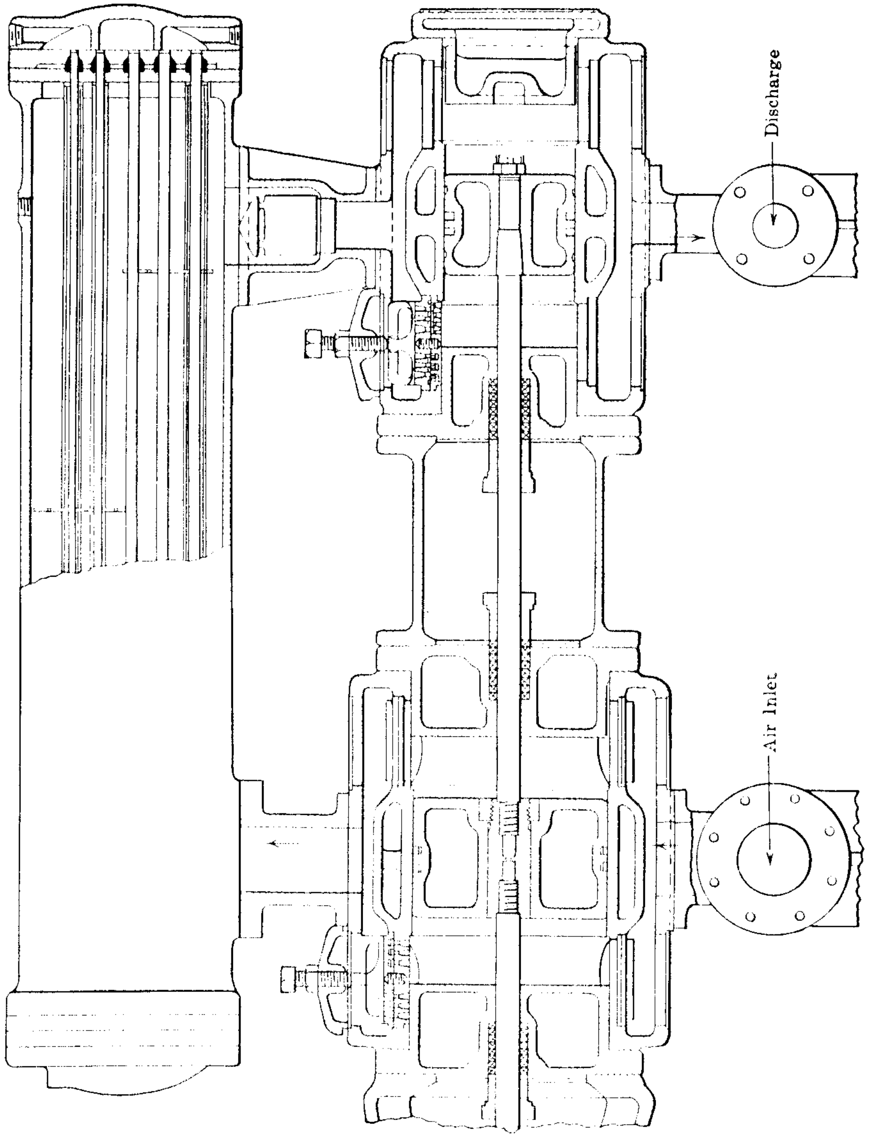

FIG. 29.—Norwalk Co. Straight-line, Two-stage, Belt-driven Compressor, Type TB-S2T.

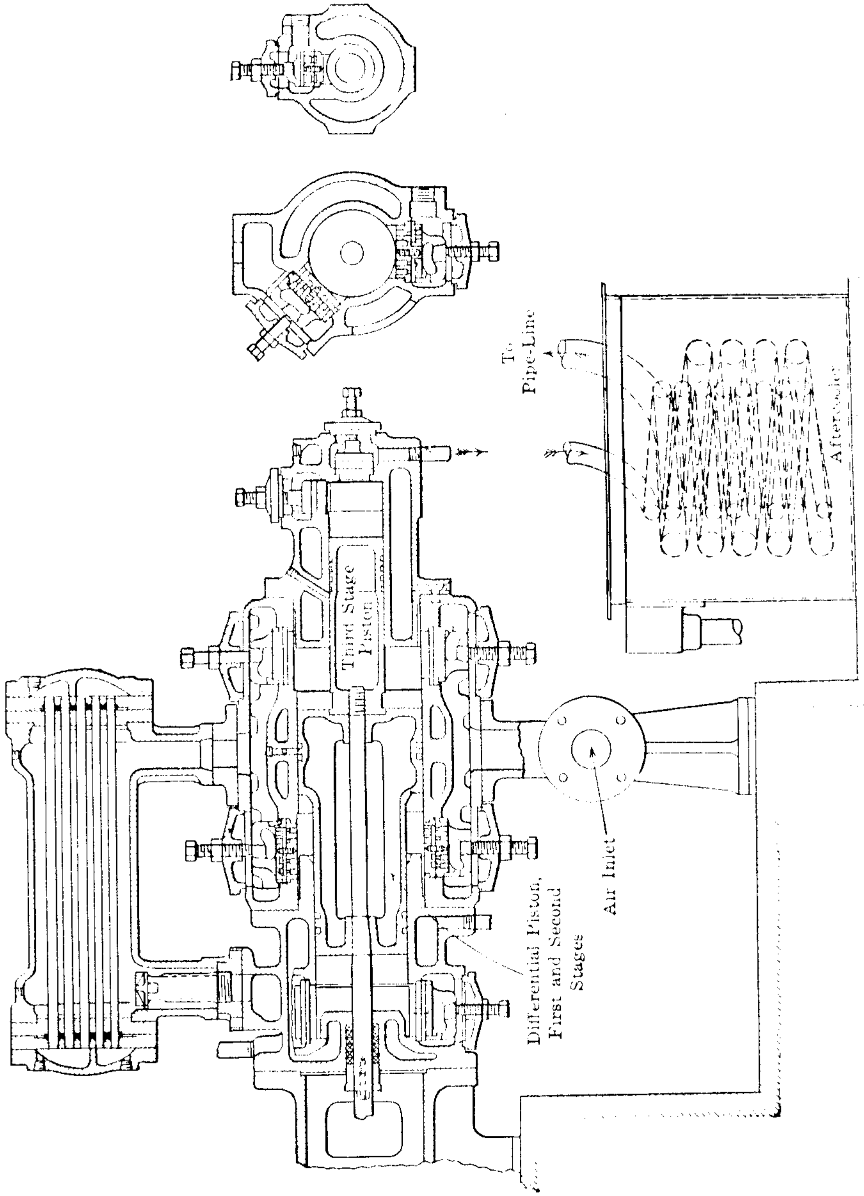

FIG. 30.—Norwalk Co. Air End of Three-stage, Belt-driven Compressor, Type TB-S3T.

Older designs of Norwalk three-stage compressors, Type AL, horizontal, with outside connecting rods, and Type VA, vertical, are now made only to order.

Four-stage horizontal and vertical compressors are also built for pressures of 2,000 to 3,000 lb., their displacements being 50 to 500 cu.ft. per min. They are driven by belt or direct-connected synchronous motor. Fig. 31 shows the Norwalk four-stage, horizontal, belt-driven Type TB-S4T. All the cylinders are single-acting. The first and second stages are produced by a differential trunk piston, in the main cylinder shell and compressing toward the power or frame end of the compressor. For the smaller sizes, the third- and fourth-stage cylinders are contained in a single secondary shell, bolted to the main cylinder; for the larger sizes, the higher-stage cylinders are in two parts, set tandem. In either case, the third-stage cylinder extends back into the hollow differential piston of the first two stages, thus affording a relatively short and rigid overall assembly.

The pistons of the third- and fourth-stage cylinders are in the form of a differential ram, with a series of packing rings. The third-stage portion of the ram is hollow; that of the fourth, solid. This ram is fitted with especial care, to prevent the serious effects of leakage of high-pressure air. All the air valves are modified poppets, of the "ring-plate" type, with powerful spiral springs (see Chapter VII, under "Poppet Valves," and Fig. 13, with accompanying text). The intercooler between the lower stages is an ordinary closed shell, as shown in Fig. 31. For the higher stages, the intercooler is of the open tank form, having coils of pipe submerged in water. This is placed below and close to the compressing cylinders, as in the lower part of Fig. 30.

Multi-stage compressors are also built by the Ingersoll Rand Co., the Sullivan Machinery Co., and some of the European makers, as noted in the list at the end of this chapter.

Pennsylvania Pump & Compressor Co. **Duplex, Two-stage, Belt-driven, Class DB-2**

Cylinder diam. in.		Stroke, in.	Rev. per min.	Displacement, cu.ft. per min.	Air pressure, lb.		Brake hp. for air pressures	
L.P.	H.P.				Min.	Max.	Min.	Max.
15	9	10	277	560	90	115	96	103
16	10	10	277	638	80	100	98	110
17	10	12	257	802	90	115	138	148
18	11	12	257	900	80	100	139	155
19	11	14	257	1,170	90	115	198	213
20	12	14	257	1,299	80	100	196	223

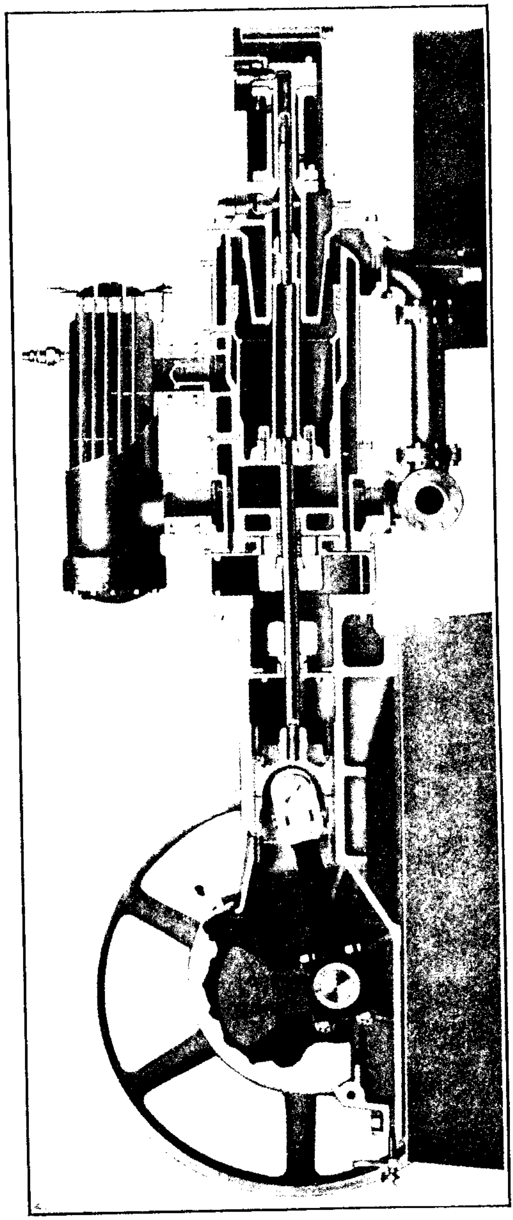

FIG. 31.—Norwalk Four-stage, Belt-driven Compressor, Type TB-S4T.

The Pennsylvania Pump & Compressor Co. also builds lines of steam- and belt-driven, two-stage compressors, with displacements of 44 to 600 cu.ft. per minute and delivery pressures ranging from 100 to 250 lb.

Reavell & Co., Ipswich, England, build eleven sizes of vertical, two-stage compressors, delivering from 500 to 5,000 cu.ft. of free air per min., compressed to 80 120 lb. (see Fig. 24). They are of two classes: motor-driven and steam-driven, the latter having compound cylinders, set tandem with the air cylinders. This company also builds two classes of three-stage compressors, for pressures of 1,000 lb. per sq.in. as follows: twelve sizes of vertical, motor- or steam-driven, with displacements of 50 to 720 cu.ft. per min., and sixteen sizes of quadruplex, motor- or belt-driven (Fig. 32), with displacements of 43.4 to 278 cu.ft. per min. The latter class is similar in general design to the single-stage quadruplex, but has two intake (low-pressure) cylinders, set horizontally, the second- and third-stage cylinders being vertical.

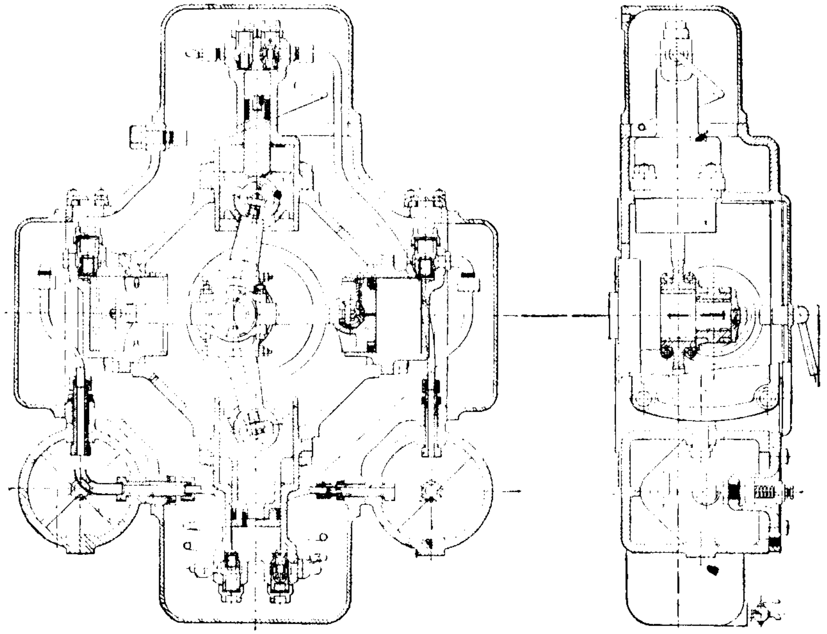

Fig. 32. Reavell & Co. Three-stage, Quadruplex Compressor.

Sullivan Machinery Co. makes five sizes of angle-compound compressors (Classes WJ-3, WN 31 and WN 32), to produce pressures of 250 to 1,000 lb. per sq.in. Sizes of cylinders, 8 and 4½ by 8 in., to 10 and 4½ by 8 in.; displacements, 186 to 299 cu.ft. per min. Those delivering the higher pressures are suitable for charging compressed-air locomotives (see Chapter XXIV).

Sullivan Machinery Co. Angle-compound, Class WJ-3, Belt-driven (Fig. 33); **WN-3, Direct-connected to Motor**

Cylinder diam., in.		Stroke, in.	Net displacement, cu.ft. per min.	Hp. for 80 120 lb.	Floor space			
L.P.	H.P.				Length, ft.	in.	Width, ft.	in.
14	8¾	10	455 620	72 100	8	10	4	0
16	9¾	10	545–752	86 120	8	11¼	4	2⅜
17	10¼	12	705–941	113 153	9	11½	5	11½
18	11	14	869 1,133	142 188	11	2½	7	6½
20	12	14	1,011–1,300	165 214	11	4½	7	10½
22	13	14	1,224–1,573	200 260	11	7	8	1
*24	14½	16	1,482 1,852	243–307	12	2	10	0

Class WN-4, Twin, Direct-connected; WJ-4, Twin, Belt-driven

14	8¾	10	910 1,240	134–214	8	3⅝	8	6
16	9¾	10	1,090 1,504	158 254	8	3⅝	8	8
17	10¼	12	1,410 1,882	210 326	9	7½	15	0
18	11	14	1,738–2,266	262 400	11	0	16	1
20	12	14	2,022 2,600	306 458	11	0	17	4⅜
22	13	14	2,448 3,146	370 554	11	0	18	3⅛
*24	14½	16	2,964–3,704	450 654	13	0	20	0

* This size is built in direct-connected units only; not for belt drive.

The exact overall dimensions of Classes WN-4 and WJ-4 depend upon type of electric motor used.

A straight-line, horizontal, two-stage, belt-driven compressor (Class WH-6) was formerly built by this company, but it now recommends in its place the angle-compound. In the angle type the moving parts are well balanced, tending to smoothness of operation. For their inertia diagrams, see Chapter IV, Fig. 2 and accompanying text.

A recent (1929) design of the Sullivan angle-compound compressor, built in nine sizes, is direct-connected to a Diesel oil-engine. These compressors are fitted with "wafer" air valves (see Fig. 16, Chapter VII). The advantages and wide range of service of oil-driven compressors are outlined on page 84.

Walker Bros. build the following lines of two-stage compressors: (1) horizontal, cross-compound, condensing (Fig. 34 and accompanying table); (2) a similar line of non-condensing compressors; (3) a line having the same size air cylinders as (1) and (2), but with the steam end proportioned for 120 and 140 lb. pressure; (4) a line having the same air end as (3), but non-condensing steam end; (5) condensing and non-

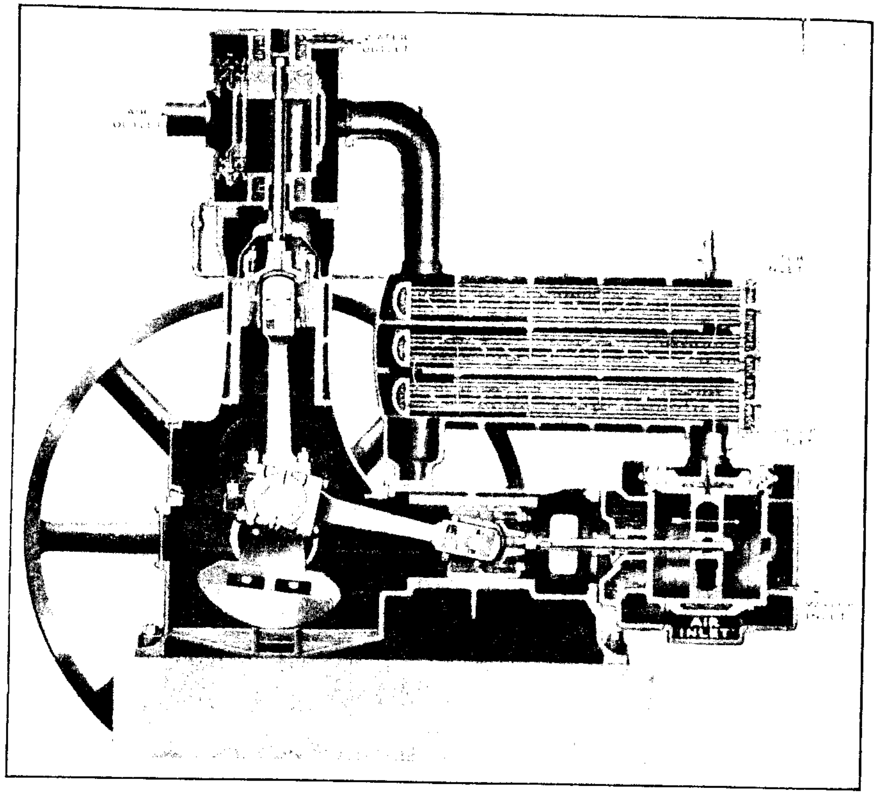

FIG. 33.—Sullivan Machinery Co. Angle-compound Compressor, Belt-driven, Class WJ 3.

Sullivan Machinery Co. Angle-compound, Direct-driven by Diesel Engine

Air cylinders, in.			Net displacement, cu.ft.*	Overall, in.			Air pipes, in.	
Diameter		Stroke						
L.P.	H.P.			Length	Width	Height†	Inlet	Disch.
12	7½	8	313-418	82	46	80½	4	2½
14	8¾	10	455-620	86	44	90	8	5
16	9¾	10	545-752	86	44	90	8	5
17	10¾	12	705-868	98	47	101	8	5
18	11	14	869-1,052	113	50	119	10	5
20	12	14	1,012-1,300	113	50	119	10	6
22	13	14	1,224-1,573	113	51	119½	10	6
26	15½	16	1,760-2,200	......	......	122	......	......
28	17	16	2,043-2,553	......	......	122	......	......

* Omitting area occupied by piston rod

† Above bottom of bed plate

condensing compressors of from 1,100 to 9,400 cu. ft. capacity, for steam pressures of 100, 120 and 140 lb. and 60-80 lb. air pressure; (6) power-driven compressors with air capacities of 500 to 3,000 cu. ft. per min., at 80 and 100 lb. pressure; (7) vertical, steam-driven, high-speed compressors (Fig. 35). This firm makes a specialty of large, horizontal compressors.

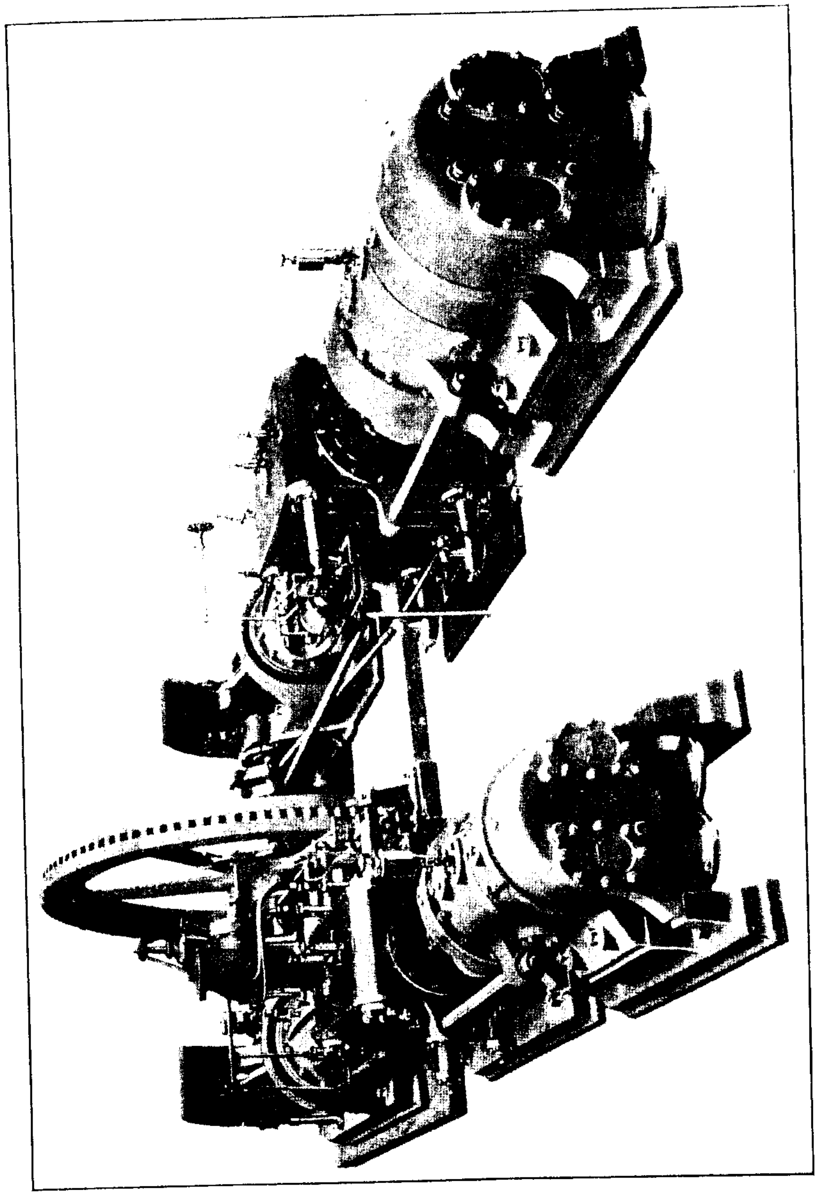

FIG. 34—Walker Bros. Corliss Cross-compound, Two-stage Compressor.

Walker Bros. **Horizontal, Two-stage, Corliss Cross-compound, Condensing** (Fig. 34)

Steam and air pressures, 100 lb.

Cylinder diam., in.				Stroke, in.	Rev. per min.	Capacity, cu.ft. free air per min.	Floor space	
Steam		Air						
H.P.	L.P.	L.P.	H.P.				Length, ft. in.	Width, ft. in.
13	24	20	12	30	100	1,000	25 3	11 0
15	26	22	13	30	100	1,250	26 10	12 2
16	28	24	15	36	90	1,660	30 3	13 8
17	31	26	16	36	90	1,880	30 3	13 8
18	33	28	17	39	85	2,200	34 5	15 0
20	36	30	18½	42	80	2,600	34 6	15 6
21	38	32	20	42	80	3,000	34 6	15 6
22	40	34	21	42	80	3,350	37 4	16 0
23	42	36	22	42	80	3,750	38 2	16 0
24	44	38	23	48	70	4,200	11 4	17 6
26	47	40	24	48	70	4,650	41 4	17 6
27	49	42	26	54	65	5,300	44 0	17 7
29	52	44	27	54	65	5,850	44 0	17 7
32	57	48	29	60	60	7,100	50 5	20 3
34	61	52	32	60	60	8,100	52 0	22 0

Worthington Pump & Machinery Corp. **Cross-compound, Two-stage** (Fig. 36)

Initial steam pressure, 110 150 lb., air pressure, 100 lb.

Cylinder diam., in.				Stroke, in.	Rev. per min.	Displacement, cu.ft. per min.	Floor space		Air pipes in.	
Steam		Air								
H.P.	L.P.	L.P.	H.P.				Length, ft. in.	Width, ft. in.	Inlet	Disch.
9	15	13	8	10	265	402	12 11	7 6	6	4
10	17	15	9½	12	225	519	13 10	7 9	7	5
12	20	17	10½	14	225	823	15 3	8 3	8	6
14	22	20	12	14	225	1,140	16 0	9 0	10	6
14	23	22	13	16	200	1,100	17 3	9 6	10	7
16	25	23	14	16	200	1,531	18 0	10 3	12	7
16	25	24	14	18	185	1,735	18 9	11 0	12	7
17	29	26½	16	18	180	2 040	20 9	11 0	12	8

Duplex, Two-stage, with Simple Steam End

Steam pressure, 125 150 lb., air pressure, 100 125 lb.

12	16	9½	14	245	790	17 0	8 6	7	5
13	18	11	16	220	1,025	18 9	9 3	8	6
14	20	12	16	220	1,270	19 0	9 3	10	6
15½	22	13	18	200	1,571	21 6	10 0	10	7
16½	23	14	18	200	1,721	21 6	10 3	10	7
18½	26½	16	21	170	2,265	23 0	11 0	12	8
20	29	18	24	165	3,010	26 0	12 0	14	8

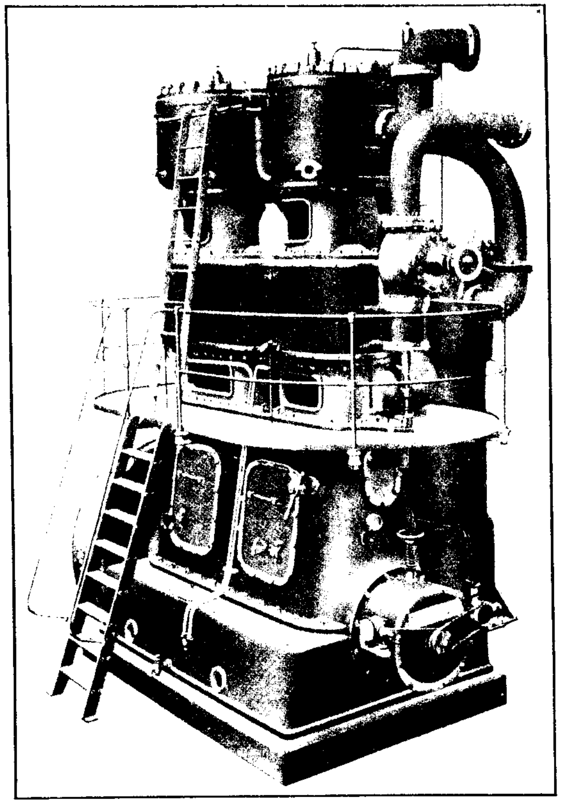

FIG. 35.—Walker Bros. Vertical, Two-stage Compressor, with Compound Steam Cylinders.

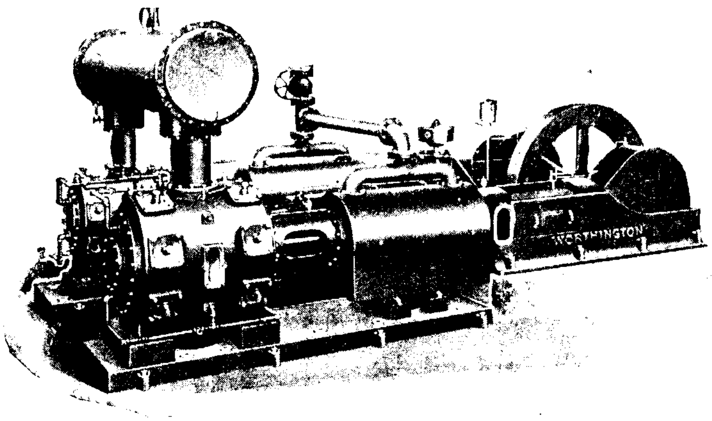

FIG. 36.—Worthington Pump & Machinery Corp. Cross-compound, Two-stage Compressor.

Worthington Pump & Machinery Corp. Duplex, Two-stage, Belt-driven (Fig. 37)

Cylinder diam., in.		Stroke, in.	Rev. per min.	Displacement, cu.ft. per min.	Motor hp.	Average floor space, including motor		Air pipes, in.	
L.P.	H.P.					Length, ft. in.	Width, ft. in.	Inlet	Disch.
12	7½	9	300	350	60	9 6	8 6	5	3
13	8	10	300	455	75	10 0	9 0	6	4
15	9½	12	277	670	125	10 6	9 6	7	5
17	10½	12	257	806	150	11 0	10 0	8	5
18	11	14	257	1,050	175	12 0	10 6	8	6
20	12	14	257	1,300	225	12 6	11 0	10	6
22	13	16	225	1,570	250	11 0	12 0	10	7
23	14	16	225	1,720	300	14 6	12 0	12	7
26½	16	18	200	2,290	400	16 0	12 6	12	8
29	18	24	180	2,874	500	17 6	13 6	14	10
33	20	24	164	3,865	650	19 0	15 0	16	10

Besides the types of compressor listed in the accompanying tables, the Worthington Corporation builds a line of duplex, two-stage compressors, driven by synchronous motors and provided with variable-capacity, five-step control (Fig. 38). For details of this mode of control, see latter part of Chapter VIII.

Oil-engine Drive for Compressors has been referred to in connection with Figs. 27 and 28, and the accompanying tables of specifications. Since oil is a concentrated form of fuel, it will bear a much higher transport cost than an equivalent amount of coal for a boiler plant. Therefore, oil-engine drive is well adapted to isolated compressor plants in remote districts, or in dry, desert areas where coal and water are costly or unobtainable, and electric power is not available.

For some years after this form of drive was first used for compressors, it was looked upon as applicable only to the above exceptional conditions. It is now recognized, however, as useful for a wider range of service: (*a*) field work of semi-permanent character, requiring more air than can be furnished by compressors of portable type; (*b*) where oil-engines are already employed for other power purposes; (*c*) wherever it is necessary or desirable to avoid the installation of boilers and their accessories.

Comparative estimates will generally show that oil-driven compressors are economical in both first cost and operating charges. The Ingersoll-Rand Co. states that, where fuel oil costs 6 cents per gallon, a 600-cu.ft. unit will operate at a fuel cost of 36 cents per hour; that is, 1,000 cu.ft. of free air will be delivered at 100 lb. pressure at a cost of 1.5

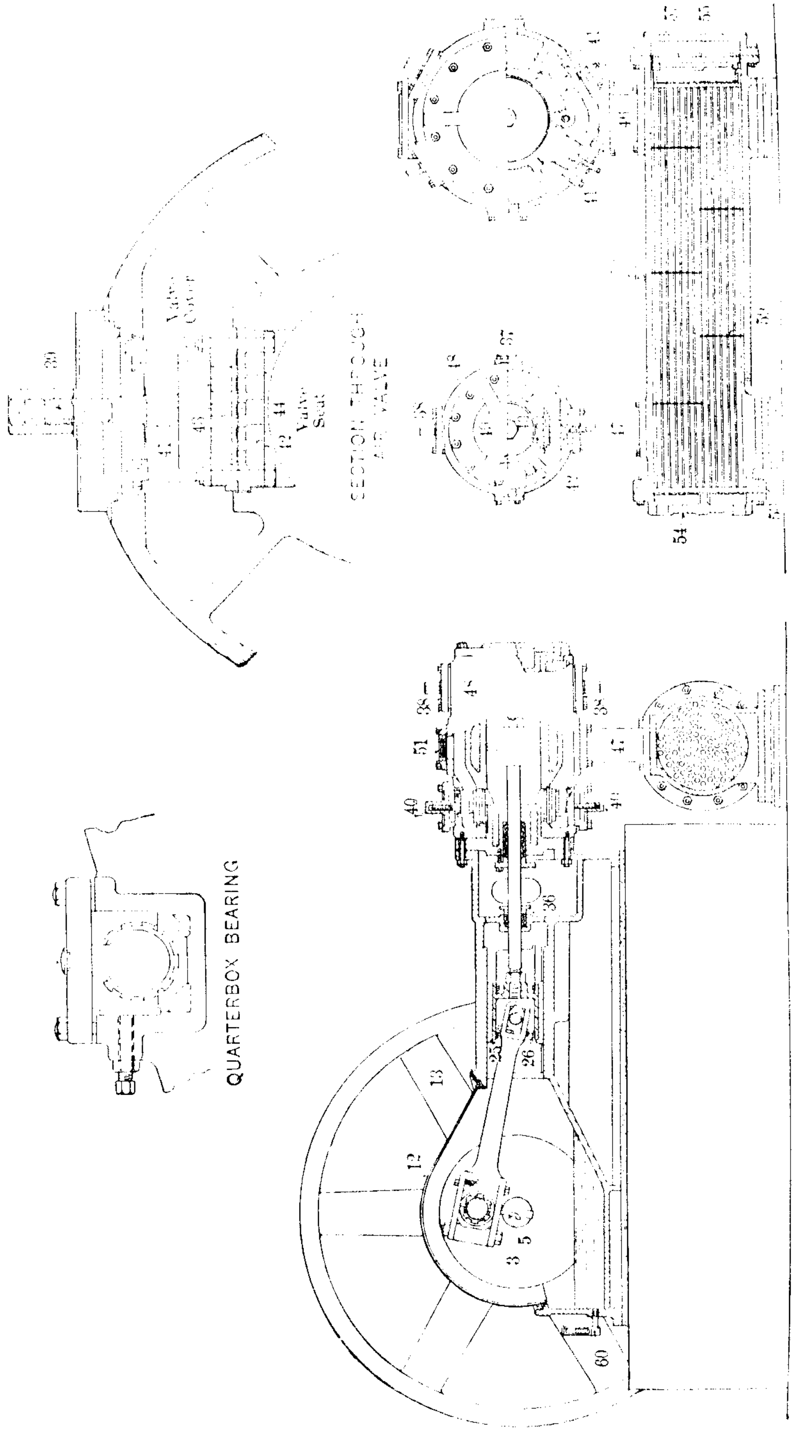

FIG. 37.—Worthington Pump & Machinery Corp. Duplex, Two-stage, Belt-driven Compressor.

cents. Similar figures are given by the Sullivan Machinery Co. (see specifications of its Diesel-engine-driven compressor (p. 80).

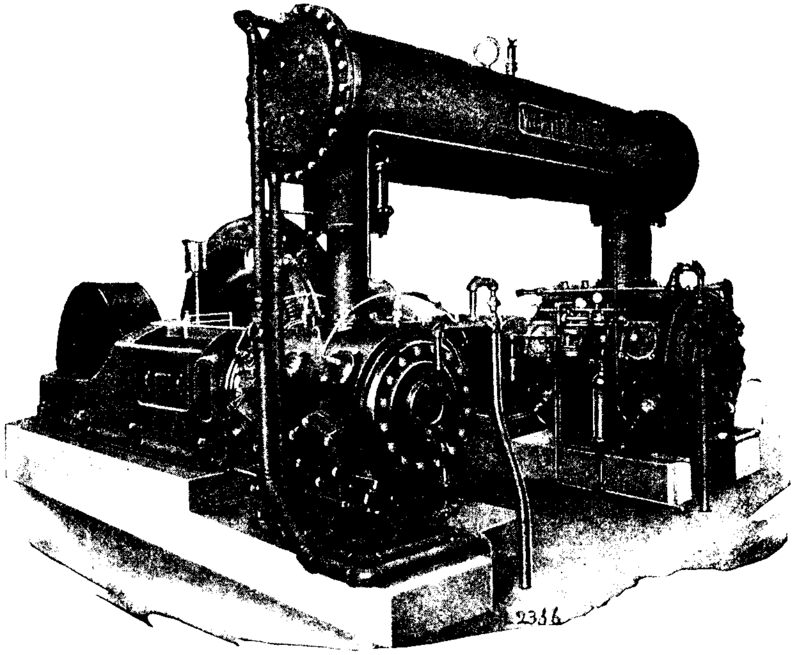

Fig. 38.—Worthington Pump & Machinery Corp. Duplex, Two-stage Compressor, with Direct-connected Motor.

Portable Compressors

These, made by nearly all compressor builders, are generally driven by gasoline engines, sometimes by electric motors. They are designed for service of temporary or semi-permanent character, like surface rock excavation, road making, and running riveting hammers; also, for mine prospecting operations, shaft sinking and sundry underground uses. They comprise two classes:

A. Compressors mounted on heavy trucks, with steel or rubber-tired wheels (or on skids), for surface operation only. Some move by their own power; others are on trailer trucks.

B. "Mine-car" compressors, for underground service. Having flanged wheels, for running on mine tracks, they readily move from place to place, in following drilling work or for operating small, temporary compressed-air pumps.

Following are examples of truck-mounted portable compressors, illustrating differences in design. All of them deliver air at a pressure of 100 lb., except where otherwise noted.

Air cylinders (2), in.		Rev. per min.	Dis-placement, cu.ft. per min.	Engine cylinders (4), in.		Net weights, lb.		
Diam.	Stroke			Diam.	Stroke	Steel wheels	Trailer	Skids
Chicago Pneumatic Tool Co (Fig. 39)								
5¼	5	800	100	4	5	4,100	4,500	3,710
6¼	6	750	160	4¾	5¼	6,900	7,900	6,175
8	6	630	220	6	7	8,850	9,750	7,800
8¼	8	600	300	6½	8	9,600	10,850	8,500
Etablissements François (Fig. 40)								
6.69 1 cyl.	5.1	750	77	4.33 2 cyls.	5.9	2,640	....	...
6.69 2 cyls.	5.1	750	154	4.33 4 cyls.	5.9	4,015	....	...

These are also built with electric motor-drive, and as trailers.

Air cylinders (2), in.		Rev. per min.	Dis-placement, cu.ft. per min.	Engine cylinders (4), in.		Net weights, lb.		
Sullivan Machinery Co. (Fig. 41)								
2 cyls. 5½	5	800	110	4	5¼	3,365	3,665	3,110
2 cyls. 6½	5¼	790	160	4½	6	6,050	....	....
4 cyls. 5½	5	800	220	5	6½	5,360	5,700	5,100
4 cyls. 6½	5¼	750	310	5¾	7⅛	7,920	8,350	6,900

The 5½ by 5-in. sizes are also furnished with Ford truck mounting.

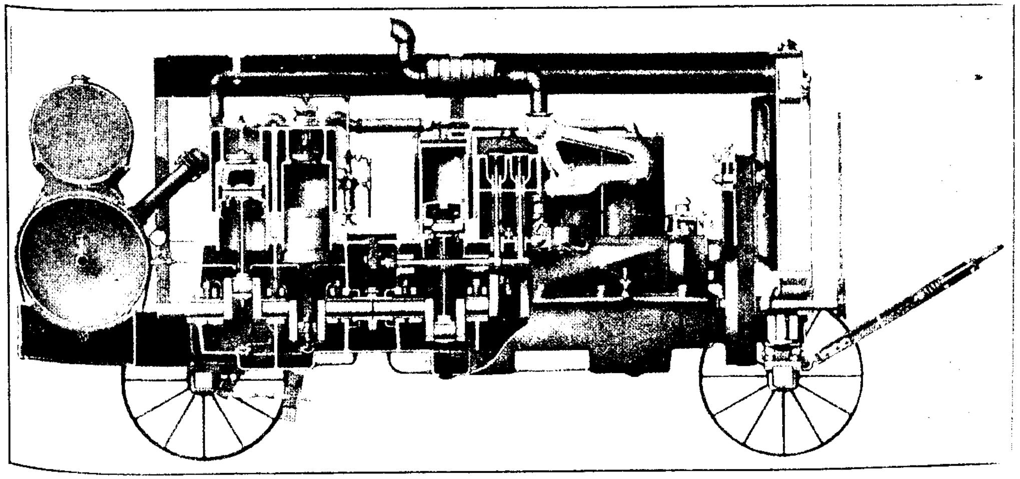

Fig. 39.—Chicago Pneumatic Tool Co. Portable, Gasoline-driven Compressor, 300 cu.ft. air per min.

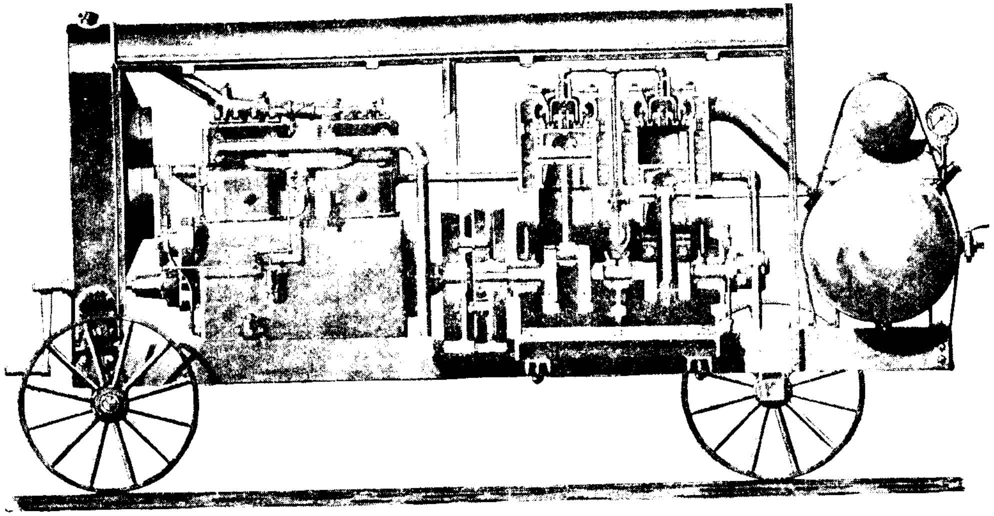

FIG. 40.—Etablissements François. Portable, Gasoline-driven Compressor.

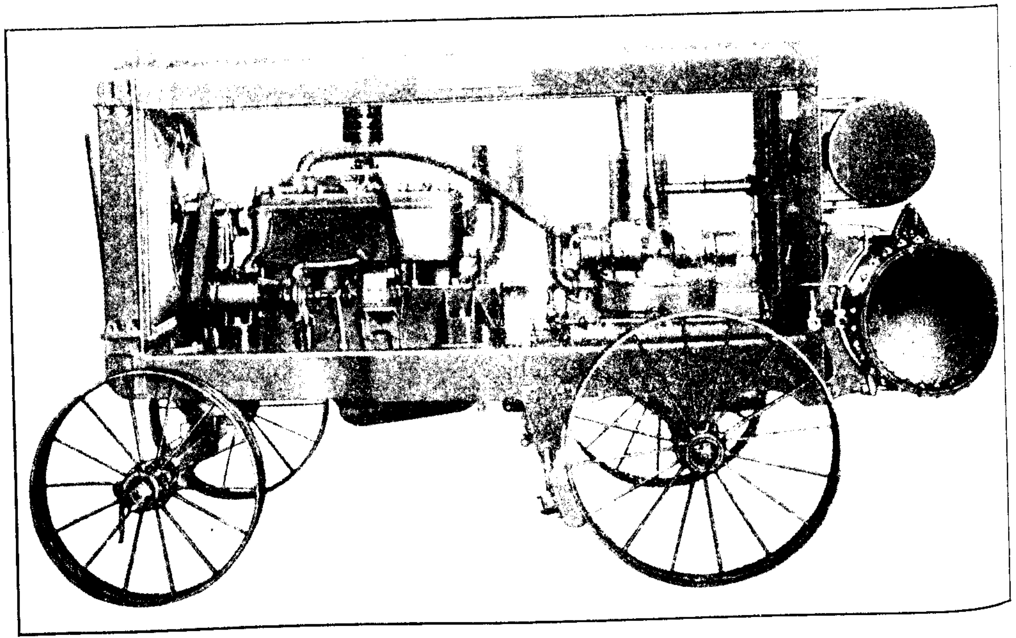

FIG. 41.—Sullivan Machinery Co. Portable, Gasoline-driven Compressor, 220 cu.ft. air per min.

Ingersoll-Rand Co. builds two sizes of portable compressor, with 4-cylinder, 4-cycle gasoline engines; air cylinders (2), 9 by 8-in. (100 lb. delivery pressure), and 10 by 8-in. (60 to 80 lb. delivery pressure). Respective weights: 6,500 and 6,000 lb. on steel wheels; 7,800 and 7,600 lb. on rubber-tired wheels.

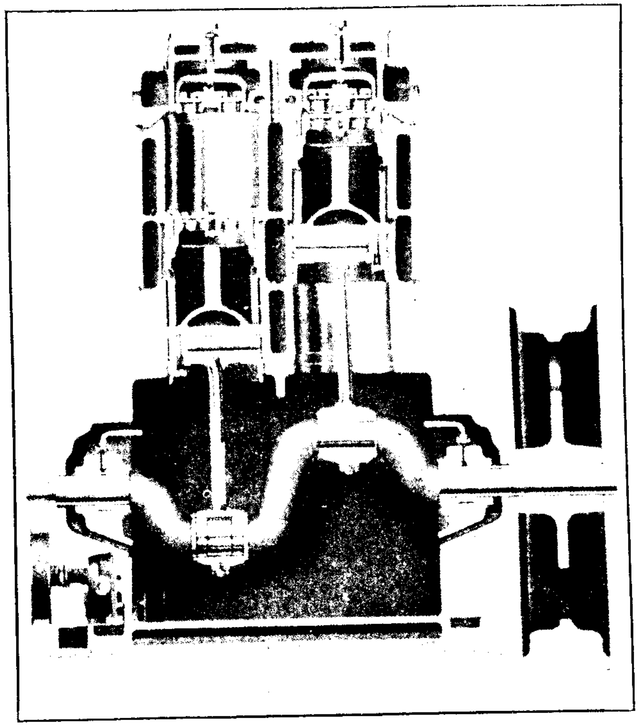

FIG. 42.—Holman Bros. Duplex, Single-acting Compressor, for Stationary and Portable Service.

Holman Bros. build three sizes of portable compressor, mounted on all-steel trucks; displacements, 82, 134 and 212 cu.ft. per min. Single-acting air cylinders: $5\frac{3}{4}$ by 6, $6\frac{3}{4}$ by 7 and $8\frac{1}{2}$ by 9 in. Maximum air pressure, 100 lb. Weights: 5,500, 6,300 and 9,800 lb.

Truck-mounted compressors for service above ground are also built by a number of other makers, including: American Air Compressor Works, Bailey & Co., Gardner-Denver Co., Pennsylvania Pump & Compressor Co., Reavell & Co. and the Rix Co.

Fig. 42 shows a duplex, single-acting compressor, for belt, motor or gasoline-engine drive. It is typical of designs adapted to high speeds of stroke, as required for portable compressors; also applicable to stationary use for small units (see also Fig. 7).

Following are examples of "mine-car" compressors, all electric-driven. The heights above rail are for 16-in. wheels.

Air cylinders, in.		Rev. per min.	Displacement, cu.ft. per min.	Air pressure, lb.	Net wt., lb.	Dimensions, in.		
Diam.	Stroke					Length	Width	Height
Ingersoll-Rand Co. Two Cylinders (Fig. 43)								
5	5	850	97	100	3,700	120	48½	43
7	6	600	160		4,650	120	60	46
9	8	390	230		7,000	126	60	53
Chicago Pneumatic Tool Co. One Cylinder (Fig. 44)								
9	8	......	89	100	......	90	58	12¼
9	8	......	144		......	90	60	12¼
12	10	......	215		......	111	65½	53¼
12	10	......	311		......	111	67½	53¼
Walker Bros. One Cylinder (Fig. 45)								
......	......	480	50–60	70	2,250	46	28	44

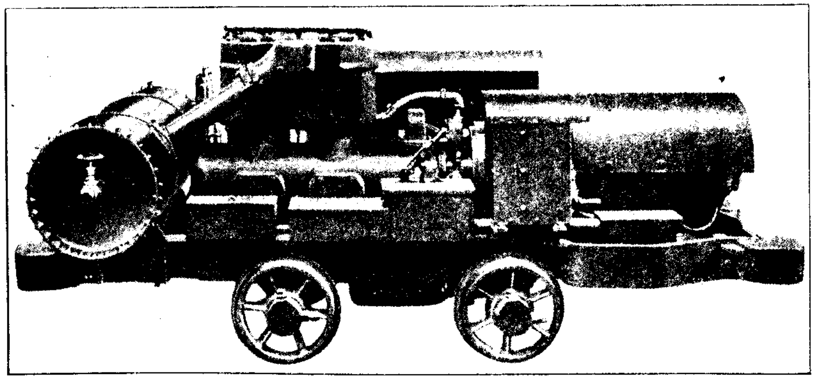

FIG. 43.—Ingersoll-Rand Mine-car Compressor, 9- by 8-in. Cylinders.

Mine-car compressors are also made in Great Britain by Bailey & Co., Holman Bros., and Reavell & Co.

Fig. 46 shows a special design of portable compressor, for use in thin coal seams, made by the Sullivan Machinery Co. To minimize the

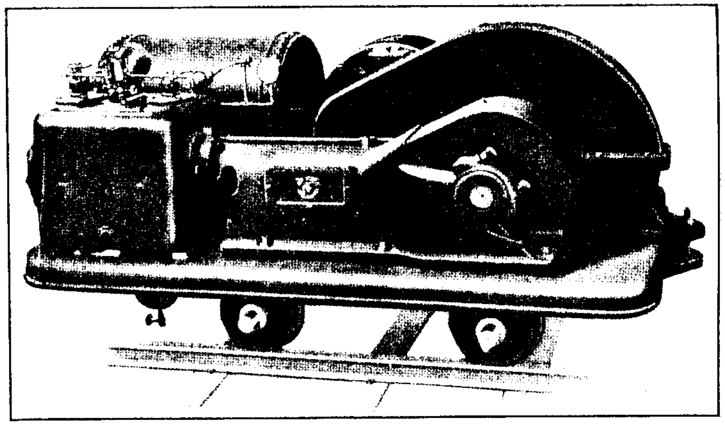

FIG. 44.—Chicago Pneumatic Tool Co. Mine-car Compressor.

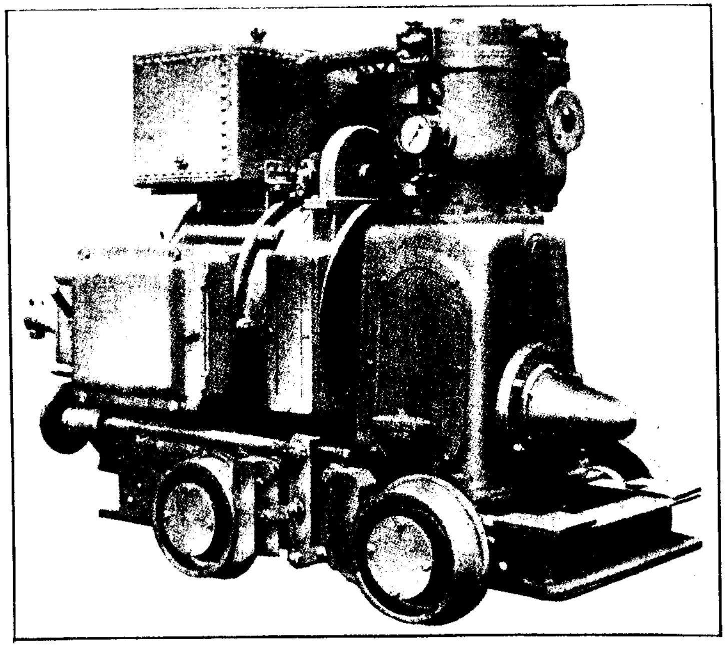

FIG. 45.—Walker Bros. Mine-car Compressor.

height above the rails, both the air cylinder and the receiver are divided into two units. The motor is "explosion-proof."

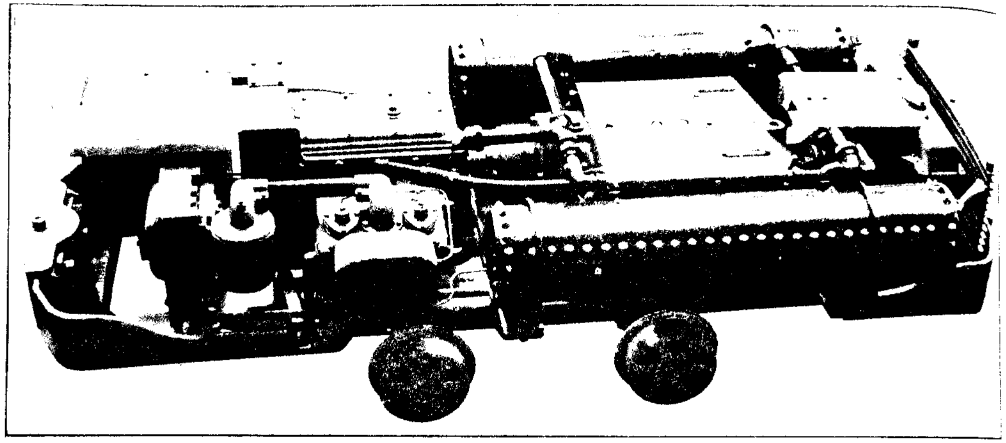

FIG. 46.—Sullivan Machinery Co. "Low-vein" Mine-car Compressor.

TURBO (CENTRIFUGAL) COMPRESSORS

The first machines of this type, based on the principles of Parsons' steam turbine, were blowing engines designed for pressures of only 15 to 30 lb. per sq.in., as required for blast furnaces, converters, etc. Rateau subsequently developed the high-pressure turbo compressor (first built by the Westinghouse Electric Co., of Great Britain), which produces air pressures of 80 to 150 lb., as suited to mine and other similar service.

Turbo compressors, in a variety of designs and sizes, are now built also by the Ingersoll-Rand Co., New York, the General Electric Co. of Great Britain (plant of the Fraser & Chalmers Engineering Works, Erith, England), the British Thomson Houston Co., Sulzer Bros., Winterthur, Switzerland, and other makers in Europe and the United States. The General Electric Co., Schenectady, N. Y., builds steam-turbine-driven, multi-stage centrifugal compressors, to serve especially as gas exhausters, etc.

In all centrifugal compressors, the air pressure produced and the power required vary directly as the square of the ratio of rotative speeds.

Driving Unit is a direct-connected steam turbine, or in some cases an electric motor. The latter is generally coupled to the compressor shaft, although gearing is employed when it is desired to provide for

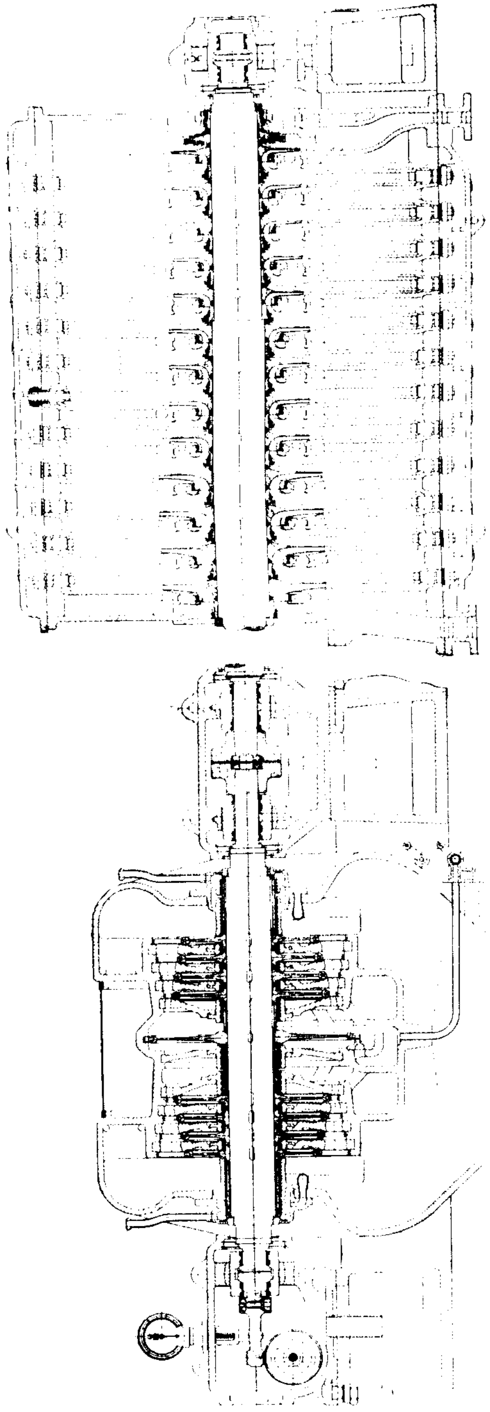

FIG. 47.—Ingersoll-Rand Co. Three-stage, Mixed-pressure Turbo Compressor. (Capacity, 10,000 cu.ft. free air per min. to 100 lb., at 4,700 rev. per min.; horse-power, 2,100.)

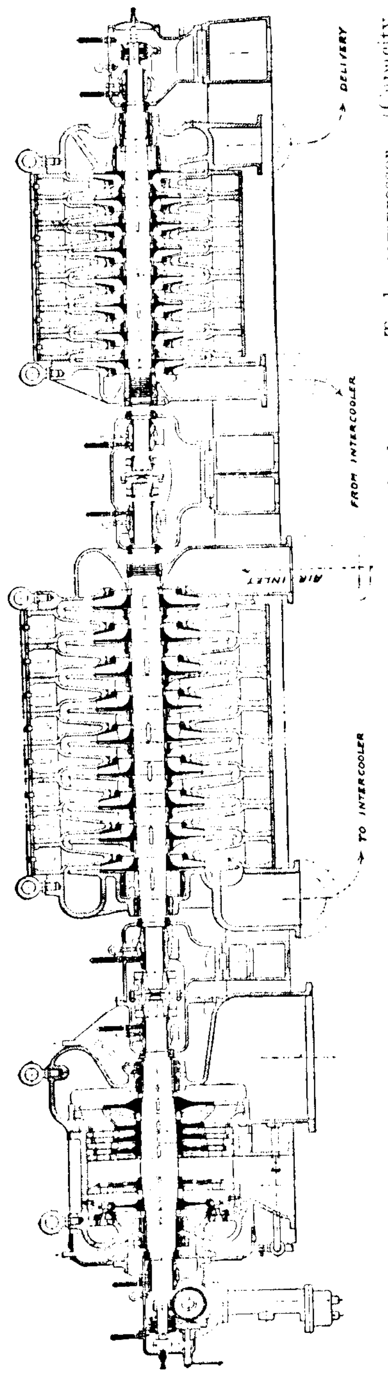

FIG. 48.—Fraser & Chalmers Engineering Works, Erith, England. Two-stage, Mixed-pressure Turbo-compressor. (Capacity, 10,000–12,000 cu.ft. free air per min. to 80 lb. (1925.)

adjusting the speed of the compressing end to meet special requirements. A steam turbine, of either the high-pressure or mixed-pressure type, is usually preferable for mine and similar service (Figs. 47 and 48). In mixed-pressure turbines, live steam may be admitted to the high-pressure (first) stage and low-pressure steam to the second stage, the low-pressure steam being the exhaust from hoisting or other engines; or, a mixture of boiler and exhaust steam may be used for both stages of the turbine. Steam turbines are provided with highly efficient governors.

The steam consumption of a typical mixed-pressure turbine is shown in Fig. 49, which gives results of a test at the New Hucknall colliery, England. In further illustration may be cited a test on a 1,000-hp., mixed-pressure turbine, driving at 4,200 rev. per min. a turbo-compressor delivering 4,100 cu.ft. of free air per min., compressed to 99 lb. Consumption of exhaust steam (taken at 15.6 lb. and discharged at 1.14 lb.) was 7.98 lb. per 100 cu.ft. free air. With the exhaust steam were used 5.05 lb. of live steam at 85 lb.

Compressing Unit.—This comprises a series of impellers, delivering air from one to another. For pressures above two or three atmospheres, compression is done in stages. Each stage consists of a set of impellers in a casing, their number per stage depending upon the required delivery pressure. The velocity at which the air issues from each impeller is converted into head (pressure) as the air flows through the passages leading to the next impeller, each adding an increment of pressure. This increment usually ranges from about 4 to 7 lb., according to the speed of rotation. The speed of turbo-

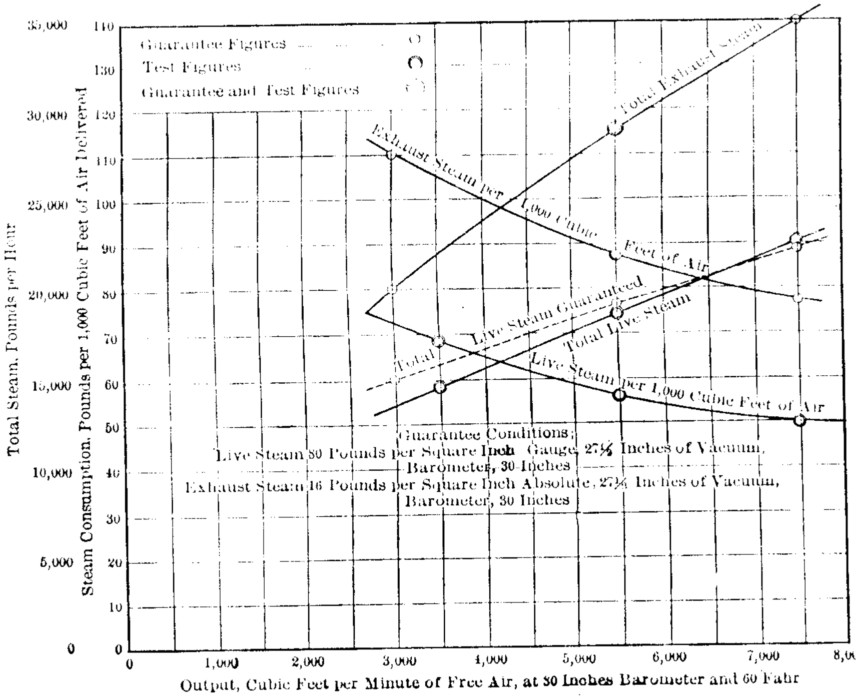

FIG. 49.—Results of Test of a British-Westinghouse Rateau Mixed-pressure Turbine and Air Compressor. Speed, 4,000 rev. per min.; output, 7,500 cu.ft. free air per min., at 80 lb. gage. Test made by W. F. Mylan (*Trans.* Instn. Min. Engs., England, Vol. 45, pp. 245–264).

compressors is commonly 3,000 to 4,500 rev. per min., the velocity of flow in the impeller passages being 300 ft. per sec. or more.

Fig. 48 shows a two-stage turbo, the first stage of nine impellers, second stage, eight impellers; each set being inclosed in a separate casing, with a coupling and bearings between them. Fig. 47 shows a three-stage turbo, the stages being in divisions of a single casing; the first and second stages have four impellers each, the third stage five impellers. A series of impellers in perspective, without their casing, is shown by Fig. 50.

Water-jackets are applied to the spaces between the exterior casing and the casings of the individual impellers. In the first stage, where the air pressure is low, the temperature rises rapidly, notwithstanding the water-jackets; but, as the density of the air under compression increases, cooling becomes more effective. The volume of cooling water required, at 70° F., is roughly 165 gal. per min. per 1,000 hp. Some engineers estimate 3,000 gal. per hr. per 1,000 cu.ft. of free air compressed per min. to 100 lb. On leaving the casing of the first stage, the air passes through an intercooler to the second stage, and so on, if there are more than two stages. For details of intercoolers, which are of the surface-condenser type, see Chapter VI.

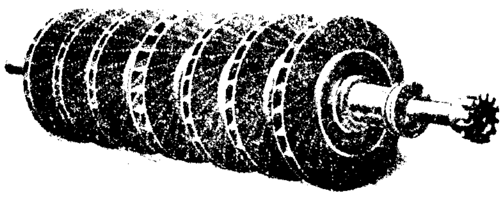

FIG. 50.—Series of Impellers, Westinghouse Turbo-compressor.

The final air temperature for a delivery pressure of 85–115 lb. ranges from say 130° to 170° F., depending upon the temperature of the cooling water. In a recorded case of a three-stage turbo, with cooling water at 50° F. and delivery pressure of 120 lb., the final air temperature was 136° F.

Examples of Turbo-compressors

Number of stages	Rev. per min.	Cu.ft. free air per min.	Delivery pressure, lb.	Hp	Where used
2	4,200	4,400	99	1,000	
4	3,000	20,000 to 23,000	135–170	4,000	South Africa
4	3,000	58,800	140–170	12,000 to 13,000	South Africa
3	4,700	10,000	100	2,100	....
3	....	6,000	80–100	....	British colliery
3	....	2,000	100	....	Factory
3	....	30,000	95	....	South Africa
2	4,000	7,500	80	1,480	British colliery

In general, turbos compressing 5,000 to 12,000 cu.ft. of free air per min. to 85–120 lb. should run at say 3,500–3,800 rev. per min. Smaller sizes, down to 2,500 cu.ft. of air compressed to 70–85 lb., and 1,750 cu.ft. to 60 lb., should be designed for 4,200–4,500 rev. per min.

Low-pressure turbos (blowers) are mainly for furnishing blast for smelting furnaces, Bessemer converters and other metallurgical service. Their pressures usually range from about 5 to 20 lb. per sq.in., sometimes as high as 30 or 35 lb.; capacities, 6,000 to 45,000 cu.ft. of free air per min., with at least one example of 60,000 cu.ft. compressed to 30 lb. The chief differences between these and high-pressure turbos are in the number of impellers, the use of stages and the provision for dealing with the heat of compression. Single-stage centrifugal blowers, for supplying low-pressure blast for cupola furnaces (0.75 to 3 lb. per sq.in.), are built by the General Electric Co., Schenectady, N. Y., Ingersoll-Rand Co. and other makers.

For the gold mines on the Witwatersrand, South Africa, the Victoria Falls and Transvaal Power Co. has installed a number of large turbo-compressors. Compressed air is transmitted from central stations to the individual mines, through many miles of piping. These turbo-compressors are of three and four stages, the largest having capacities of nearly 60,000 cu.ft. of free air per min.; delivery pressure, 140 to 170 lb. per sq. in.; horsepowers, 12,000 to 13,000. At one station there are twelve 4,000-hp. turbos, each delivering 20,000–23,000 cu.ft. free air per min., compressed to 135–170 lb., at an efficiency of 67.5 per cent (referred to isothermal). The latter are four-stage: two low-pressure, one intermediate and one high-pressure, each stage comprising a series of impellers; speed, 3,000 rev. per min.; driven by synchronous motors. Other turbos of this company are driven by steam turbines, using superheated steam.

Efficiency of the compressing end of large turbos should range from say 65 to 80 per cent, at full load, referred to the isothermal curve. Turbo-compressors having capacities exceeding 6,000 cu.ft. free air per min. generally work at the higher efficiencies; but a turbo having 70 per cent efficiency at full load may show only 50 per cent efficiency at half load.

Air Governors for turbo-compressors are designed to maintain a nearly constant pressure under quite wide fluctuations in the volume of air used (see end of Chapter VIII). For example, a 5,000-cu.ft. Thomson-Houston turbo, at a normal speed of 4,600 rev. per min., showed a variation of only 2 or 3 lb. pressure for outputs ranging from 1,500 to 6,000 cu.ft. free air per min.

Field of Use.—For mine service, in point of first cost, it is generally not advisable to use turbo-compressors for outputs smaller than 2,000–2,500 cu.ft. free air per min., nor for pressures much exceeding 75–85 lb. For smaller outputs or higher pressures, reciprocating compressors are preferable. For central power stations, however, like those in the

Transvaal (cited above), containing groups of large capacity units, turbos have been found satisfactory for high air pressures.

MISCELLANEOUS COMPRESSORS

Rotary Compressors of several types have appeared, most of them being of small capacity. As they involve the use of pistons, and are capable of producing pressures to 100 lb. or more, such as are suitable for machine drills and other mine service, they should not be confused with low-pressure rotary blowers, noted previously under Turbo-compressors.

A recent successful design of rotary compressor is the Hele-Shaw-Beacham, built by Holman Bros., Camborne, England, which promises to be quite widely adaptable. Single-stage compressors of this make have three cylinders, set radially 120° apart; the two-stage have two low-pressure and two high-pressure cylinders, set at 90°. Fig. 51 shows a two-stage compressor, of 300 cu.ft. per min. displacement. Following is a list of the sizes of stationary compressor of this type now (April, 1929) being built:

Displacement, cu.ft. free air per min.		Rev. per min.	Cylinders, in.		Number of cylinders		Delivery pressure, lb.
			Diam.	Stroke			
Single-stage	58	2,500	4	$1\frac{1}{16}$		3	60
	77	1,500	5	1 5		3	100
	125	1,500	6	$1\frac{11}{16}$		3	100
	162	1,500	6.5	$1\frac{7}{8}$		3	100
Two-stage	106	1,000	7 and $4\frac{3}{8}$	$2\frac{3}{8}$	Per stage	2	100
	200	1,000	$9\frac{1}{4}$ and $5\frac{3}{8}$	$2\frac{9}{16}$		2	100
	300	1,000	$10\frac{1}{2}$ and $6\frac{3}{8}$	3		2	100

The cylinders are parts of a single casting, which rotates on ball bearings around a central, non-rotating cylindrical valve, the whole being inclosed in a cylindrical casing. The cylinder body is rotated by a hollow central shaft, driven by an electric motor. One end of the valve is carried on roller bearings within the driving shaft, the other end being keyed to the compressor casing. Ports and passages in the valve admit air to the low-pressure cylinders, whence it passes through an intercooler, thence back through the valve to the high-pressure cylinders. The pistons are carried on roller bearings attached to their gudgeon pins, these rollers engaging with two "floating rings," which are in ball-bearing contact with the outer, fixed casing. The axes of the rings

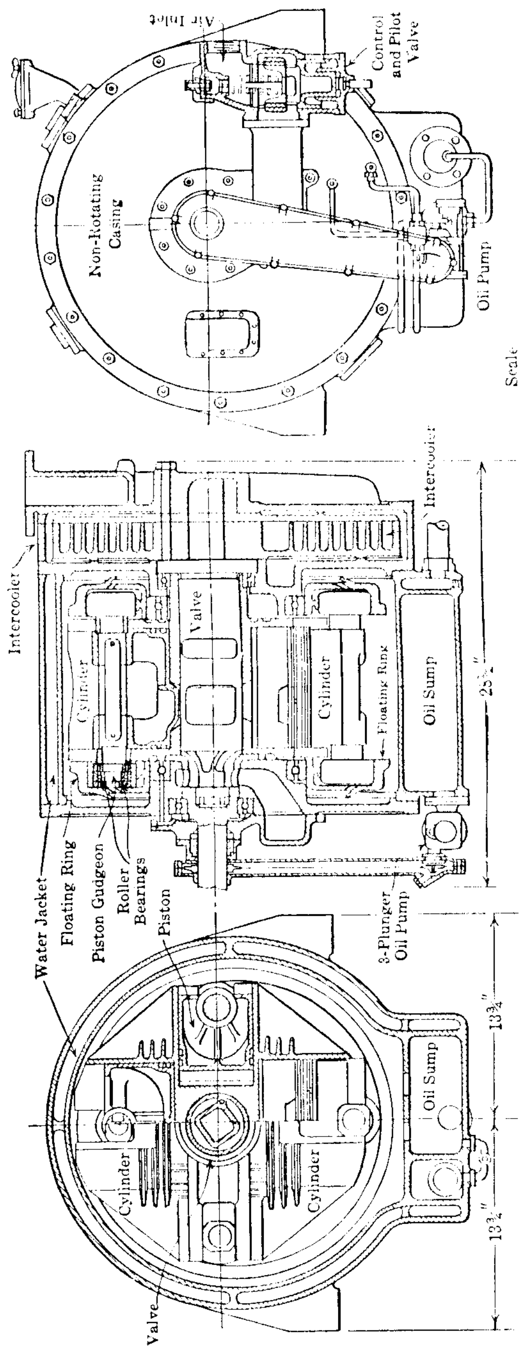

FIG. 51.—Hele-Shaw-Beacham Four-cylinder, Rotary, Two-stage Compressor, built by Holman Bros.

are eccentric to the axis of the valve and central driving shaft; hence, when the cylinder body rotates, a relative reciprocating motion takes place between the pistons and cylinders. The eccentricity of the rings equals one-half the stroke of the pistons. On one side of the casing is the intercooler, a thick aluminum disk, water-cooled on each side and containing channels through which the air passes.

Pistons, rings and valve are lubricated by a small three-cylinder pump, which injects oil into the intake air. In the original design the oil was extracted from the compressed air by an "eliminator," attached to the compressor casing. This consisted of a baffle-plate, having ⅛-in. holes, through which the air passed to a nest of vertical wires. The particles of oil caught on the wires were collected in a sump, whence the oil was pumped to the floating rings, and to the central valve to form an oil seal. Oil sealing of the valve is now (1929) effected by natural leakage of oil from the cylinders. Tests have shown that the quantity of oil carried by the discharge air is negligible.

A practically perfect dynamic balance of the moving parts is attained by equalizing the weights of the low-and high-pressure pistons, gudgeons and rollers. This is done by making the high-pressure pistons of iron and the low-pressure pistons of aluminum alloy, and permits high rotative speeds. For further details of this compressor, as originally designed, see Trans. Instn. Min. & Met. (London), Vol. XXXV, p. 102. Several changes in the design were made in 1927 and 1928.

Among other rotary compressors are those of Reavell & Co., Ipswich, England. They are driven by belt or direct-connected motor; delivery pressures, 5 to 20 lb. per sq.in.; rotative speeds, 600 to 3,000 revs. per min.; displacements, 24 to 2,170 cu.ft. per min.

Water-driven Compressors are generally of special designs. Impulse wheels, like the Pelton or Knight, are well adapted to this service, and are best direct-connected, by mounting them on the compressor crank-shaft. Their diameter can be adjusted to give the required speed of stroke for the compressor, while still permitting a sufficiently high peripheral velocity of the wheel to insure economical use of the water. For best efficiency, the peripheral velocity of an impulse wheel is theoretically one-half the velocity of the jet from the nozzle. Since heavy heads of water involve high peripheral velocities, direct-connected wheels are sometimes of great size (Fig. 52). A small-diameter wheel for a high head would require belting or gearing, which would be mechanically undesirable.

If there is a sufficient volume of water, low heads may be utilized by employing multiple nozzles, directed tangentially at two or more points of the periphery of the wheel. The flow of water may be regulated

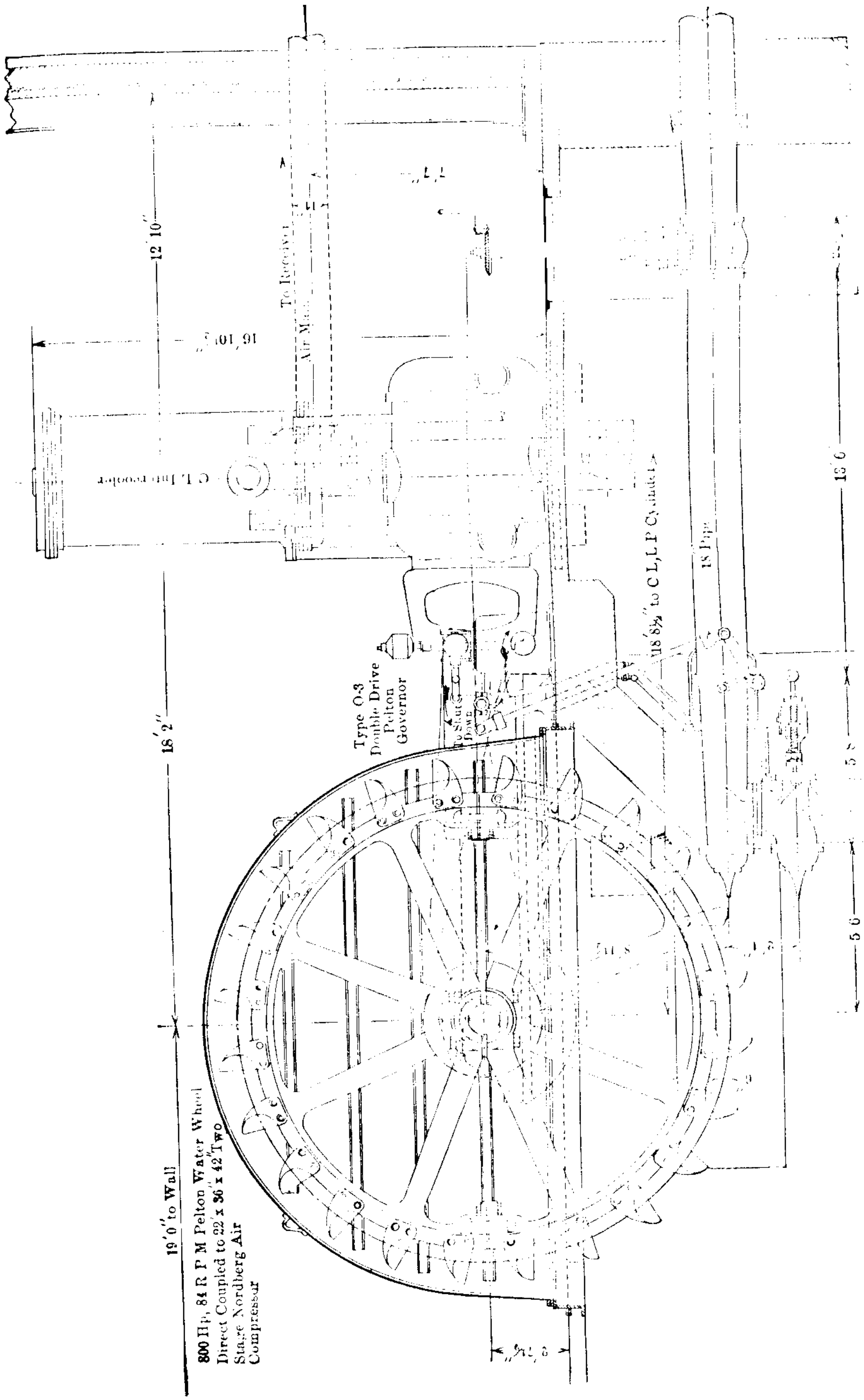

FIG. 52.—Nordberg Compressor, Direct-connected to a Pelton Wheel.

by an ordinary gate valve; but for high heads a slow-moving gate is necessary, to avoid danger of rupturing the pressure pipe in case the compressor is suddenly stopped.

Nozzles are best of the "needle" type (Fig. 53). The axial position of the needle is adjusted by a handwheel and gearing for delivering the requisite volume of water, the needle being shaped to form a solid stream against the buckets of the wheel. The cut shows an automatic cutoff governor, controlled by air pressure from the receiver. An excess of receiver pressure, admitted to the small cylinder *a*, causes arm *b* to move backward, thus raising hood *c* in front of the nozzle and deflecting the water to the tail race, instead of allowing it to impinge on the wheel. When receiver pressure returns to normal, the hood automatically moves downward, out of line of the jet. This regulates the speed of the wheel, but does not save water by checking the nozzle flow. To economize water, a valve in the pipe from penstock to nozzle, is automatically controlled for regulating the flow.

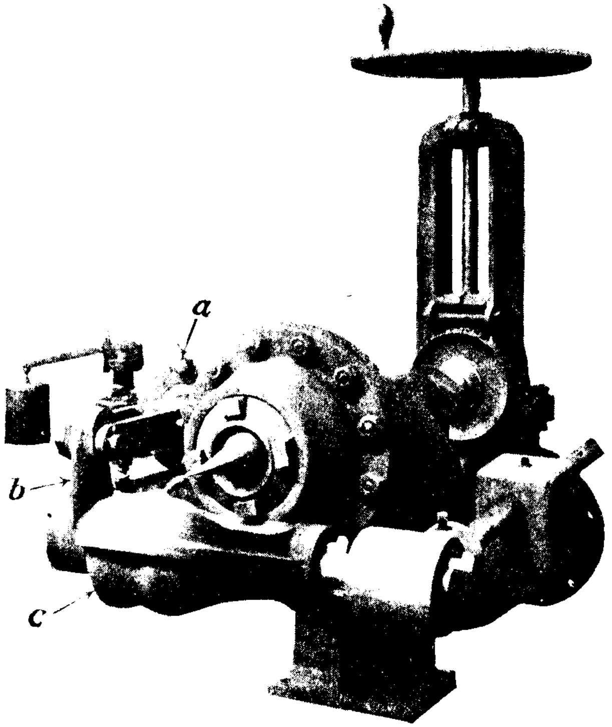

Fig. 53.—Needle Nozzle for Pelton Wheel.

Since the power developed is uniform throughout the revolution of a water wheel the compressor should be duplex, to minimize variations in torque; and the rim of the wheel is made extra heavy, to act as a flywheel. The flywheel effect of water wheels of small diameter may be increased by bolting heavy iron segments on the spokes. Fig. 54 shows a special Pelton wheel, sectionalized for mule-back transport in Mexico, the hub being designed for shrinking on forged bands, after the wheel is in place on its shaft.

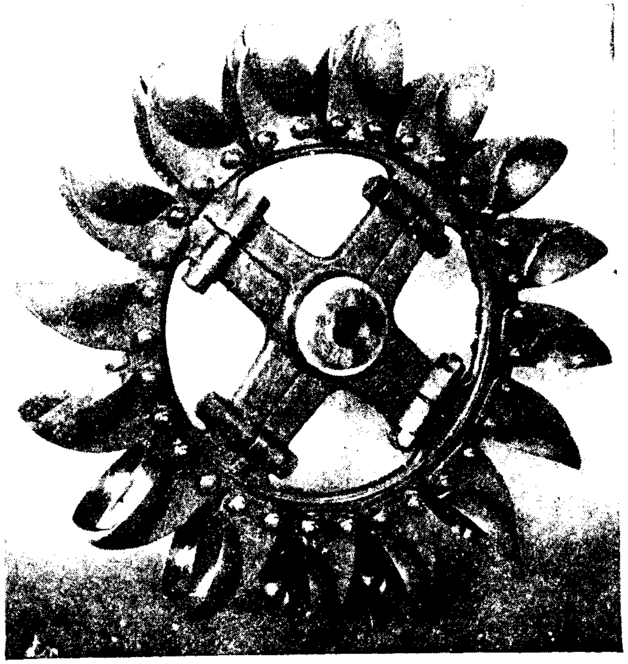

Fig. 54. Pelton Impulse Wheel.

The following details of a large plant are given in further illustration of the results obtainable from water-driven compressors.

At the Morning Mine (Federal Mining and Smelting Co.), near Mullan, Idaho, is a water-driven two-stage compressor, that has been continuously in operation since its erection in 1900. There are four cylinders, a high- and a low-pressure being set tandem on each side of a set of three Pelton wheels, mounted on the crank-shaft. Water, under a head of 140 ft., is delivered through 8,000 ft. of flume and 400 ft. of pressure pipe, driving two 12-ft wheels. Two other streams, piped respectively 1½ and 1 mile, produce heads of 1,140 and 1,420 ft.; these drive a 33-ft. Pelton wheel, placed on the crank-shaft between the smaller wheels. The 33-ft. wheel is driven by separate jets from the high-pressure lines, and, due to their difference in head, the diameter of this wheel is a mean between the diameters theoretically necessary for obtaining a peripheral velocity properly proportioned to each head. The actual mean peripheral speed is 8,000 ft. per min. To control the water under these great heads (pressures, about 190 and 610 lb. per sq. in.), slow-acting gate valves are provided, with by-passes for starting and stopping. To stop the compressor quickly the nozzles can be deflected clear of the wheel.

Each pair of air cylinders are $33\frac{1}{2}$ and 18 in. × 42-in. stroke; piston speed, 560 ft. The low-pressure cylinders compress to about 30 lb., the high-pressure to 90 lb. Inter- and after-coolers are placed in the tail-race of the smaller wheels. As originally installed, the valve gear consisted of Corliss valves for both inlet and discharge, operated by a very elaborate system of cams, eccentrics and connecting rods, to take care of seasonal fluctuations in the water supply, which caused variations in speed of the water-wheels. The Corliss valves of the high-pressure cylinders, being positively driven, gave especial trouble under lowered speed; considerable air leaked back through them, and they had to be flooded with oil to prevent them from cutting. In 1927, this complicated valve gear was replaced by plate valves (see Chapter VII), which are operating satisfactorily, with a material saving in the power that was absorbed by the old gear. Also, the compressor can now be run at a higher speed (and capacity) during periods of low water.

Indicator cards have shown this compressor to be highly efficient. An average of a number of cards gave mean pressures of: low-pressure cylinder, 17.86 lb.; high-pressure, 41.14 lb.; combined, 30.46 lb. The mean theoretical adiabatic and isothermal pressures, corresponding to the combined mean are, respectively, 36.94 and 28.5 lb. During the tests the observed temperatures were: cooling water, 38°; air at discharge from low-pressure cylinder 135°; at high-pressure inlet, 46°; high-pressure discharge, 140°; on leaving the after-cooler, 62°. Mean atmospheric temperature, 55° and of the cooling water 38°.

The following lists comprise the names of most compressor builders:

United States

Allis-Chalmers Manufacturing Co., Milwaukee, Wis.
American Air Compressor Works, Brooklyn, N. Y.
Bury Compressor Co., New York City.
Chicago Pneumatic Tool Co., Chicago, Ill.

T. H. Dallet Co., Philadelphia, Pa.
De Laval Steam Turbine Co., Trenton, N. J.
Gardner-Denver Co., Denver, Colo. (portable compressors only).
General Electric Co., Schenectady, N. Y. (turbo-compressors only).
Ingersoll-Rand Co., New York City.
Miller Gas Engine Co., Springfield, Ohio.
Nordberg Manufacturing Co., Milwaukee, Wis. (large, special compressors only).
Norwalk Co., South Norwalk, Conn.
Pennsylvania Pump & Compressor Co., Easton, Pa.
Rix Co., San Francisco, Calif.
Sullivan Machinery Co., Chicago, Ill
Vulcan Iron Works, Scranton, Pa.
Worthington Pump & Machinery Corp. (Laidlaw Works), Cincinnati, Ohio.

Great Britain

Daniel Adamson & Co., Dunkinfield, Manchester (turbo-compressors a specialty).
Alley & MacLellan, Glasgow.
W. H. Bailey & Co., Manchester.
Belliss & Morcom, Birmingham.
British Thomson Houston Co. (turbo-compressors only).
British Westinghouse Electric Co. (turbo-compressors only).
Broom & Wade, High Wickham (small compressors).
Peter Brotherhood & Co., Peterborough.
Consolidated Pneumatic Tool Co., London.
Fraser & Chalmers Engineering Works, Erith (turbo-compressors a specialty).
Fullerton, Hodgart & Barclay, Paisley, Scotland.
Globe Pneumatic Engineering Co., London.
Hardy Patent Pick Co., Sheffield (small compressors only).
Holman Brothers, Camborne.
Reavell & Co., Ipswich.
Robey & Co., Lincoln.
Seagers Limited, Dartford (specialty, gas compressors).
Tilghman's Patent Sand Blast Co., Atlantic Works, Broadheath, near Manchester.
Walker Brothers, Wigan (specialty, large horizontal steam-driven compressors).
G. & J. Weir, Glasgow (specialty, multi-stage compressors).

Continent of Europe

Atlas Diesel Co., Stockholm, Sweden.
Berliner Maschinenbau A. G., Berlin, Germany (specialty, high-pressure compressors).
A. Borsig, Tegel, Berlin, Germany.
Cockerill et Cie., Seraing, Belgium.
Dujardin et Cie., Lille, France.
Etablissements François, Bressoux-les-Liège, Belgium.
Flottmann Maschinenfabrik, Herne, Germany.
Frankfurter Maschinenbau A. G., Frankfort-on-Main, Germany.
Neumann & Esser, Aachen, Germany.
Sulzer Brothers, Winterthur, Switzerland (piston and Rateau turbo-compressors).
Zwickauer Maschinenfabrik, Zwickau, Saxony.

CHAPTER IV

COMPARISON OF COMPRESSORS AND DETAILS OF DESIGN

This chapter deals with compressors considered primarily as engines. Details of the air cylinders, including valves, water-jackets, intercoolers, air governors and unloaders, will be found in subsequent chapters.

Straight-line Compressors (Figs. 1–3, 5, 8, 11, 14, 15, Chapter III) are suitable for small or medium size units, or for temporary service, as surface excavation, shaft sinking, tunnel driving and extensive prospecting operations. Being compact and carried on a single bed-plate, they require relatively inexpensive foundations; the floor space occupied is about one-third that of a duplex compressor of the same capacity. They are built in sizes to 1,700 or 1,800 cu.ft. of free air per min. (for specifications, see Chapter III). Although useful for moderate air pressures and fairly constant loads, this type of compressor is not capable of operating with the steam economy essential for large plants; nor is it self-regulating at much less than about 40 per cent of its full load.

Duplex Compressors (Figs. 10, 16, 18, 19–21, 25, 26, 37, Chapter III), preferable to the straight-line for large units, are built in sizes to about 10,000 cu.ft. of free air per min. They are better adapted to varying loads, arising from variations of pressure and consumption of air, because the torque is more uniform. Having quartering cranks, they will run at very low speed without stopping on a center, and are self-regulating and capable of dealing economically with a range of load down to less than one-third or even one-quarter of the normal. The quadruplex, high-speed, single- and three-stage compressors (Figs. 12 and 32, Chapter III) operate at a still more uniform torque.

As a rule, the friction loss in standard duplex types, that is, the total horsepower consumed by engine friction, is no greater and may be less than that of a straight-line compressor of the same capacity. For large compressors, in good order, this loss may not exceed 5–7 per cent.* The Corliss type of engine is often used for driving large duplex compressors, as its valve gear is well adapted to dealing with variations of air pressure. Corliss valves were formerly quite largely employed for the air cylinders also; but since about 1913 they have been generally superseded by "thin-plate" or "feather" valves of a number of forms (Chapter VII).

Half-duplex compressors consist of either the right- or left-hand half of a duplex compressor, an extended crank-shaft and out-board pillow-block being provided temporarily. If a comparatively small quantity of air is needed for a time—as during the development of a mine or the sinking of a shaft—a half-duplex may be installed at first, the second half being added later. The capacity is thus readily doubled at moderate cost.

The foundation of a duplex compressor necessarily costs much more than that of a straight-line, and must be substantially built to maintain prefect alinement of the cylinders on either side of the flywheel. Foundation plans are furnished by compressor builders, so that when required the foundation, with the anchor bolts in position, can be completed in advance, ready for receiving the compressor when it arrives. The pairs of cylinders on each side are connected by trunk-frames or tie-bolts. A complete girder-frame prevents possibility of movement. (See cuts in Chapter III.)

Some mechanical engineers hold that tandem steam and air cylinders are best placed far enough apart to prevent the same portion of the piston-rod from passing alternately into each stuffing-box, because, as the piston-rod is apt to wear differently in the two stuffing-boxes, it may be difficult to keep them well packed and tight. The steam and air piston-rods are often made in separate parts, coupled between the cylinders, for convenience in dismantling the compressor for repairs, and to permit readier access to the air valves for regulating them or regrinding them to their seats. For the latter reasons, also, the steam cylinders of tandem compressors are placed nearest the flywheel.

Vertical Compressors (Figs. 17, 22–24, 35, Chapter III), generally designed for relatively high speeds, are advantageous on the score of their rigidity and in saving floor area, though economy of space is rarely of consequence at mines. As the moving parts are generally entirely inclosed, they are protected from dust, but, for the same reason, the stuffing-boxes and valves are not so accessible for adjustment and repairs as those of most horizontal compressors. The vertical type has had its greatest development in Great Britain.

Angle Compressors (Fig. 33, Chapter III), driven by belt or direct-connected electric motor, are of the two-stage form only, the low-pressure cylinder being horizontal, the high-pressure, vertical. Twin angle compressors are also built; they are economical in running expenses, and the reciprocating parts are well-balanced, giving a fairly uniform torque (Fig. 2, *c*). Their advantages as to economy of floor space and cost of foundation are similar to those of vertical compressors.

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* A test on a 300-hp. compound, two-stage, Nordberg Corliss compressor (Chapter X) showed a total efficiency of 78.1 per cent, the friction loss being 5.2 per cent.

Turbo Compressors are best applied in large units only. Since compressed air can be conveyed long distances with small losses of either volume or power, if the pipe lines are properly designed and maintained, the high-pressure turbos (Figs. 47 and 48) are particularly useful for central power stations, from which air is distributed to groups of large mines. Turbos, therefore, can not be considered as competitors in the field of work occupied by the various types of standard reciprocating compressors.

Stage Compressors.—Notwithstanding the somewhat higher friction loss in stage compressors, due to the greater number of moving parts, as compared with simple (single air cylinder) compressors, it has long been recognized that a saving in power consumption can be realized by stage compression, even for pressures of 75–100 lb., as commonly employed for machine drills. In mountain regions, where so much mining is carried on, its advantages are still greater than at sea-level (Chapter XII). The duplex form, with both steam and air ends compounded, and the electric-driven compressors exemplify the highest types. With stage compression the maximum range of steam expansion is obtainable, because the work is more nearly equalized than when done in a single air cylinder, and the compressor may be made self-regulating over its entire range of load. Thermodynamically, the efficiency of stage compression depends largely on the proper application of water-jackets to the air cylinders, the size and design of the intercoolers (Chapter VI), and on the maintenance of moderate piston speed.

The Operation of Steam-driven Compressors involves conditions that do not obtain in ordinary steam engines (see Fig. 1). At the beginning of the stroke the air in front of the piston is approximately at atmospheric pressure. As the piston advances, the pressure at first increases slowly, and then very rapidly to its maximum. The variations in the power developed in the steam cylinder, when working as usual with a cutoff, are in the reverse order. The initial steam pressure may be lower than the final air pressure, though the mean effective steam pressure is greater than the mean effective air pressure. For example, with an initial steam pressure of 60 lb., air may be compressed to 80 lb. or more. This is due to the use of heavy flywheels and reciprocating parts, which store up the surplus power in the early part of the stroke, and give it out toward the end. The consequent lack of smoothness in the running of compressors is especially noticeable in the simple straight-line type, which, when the air in the receiver is at maximum,

often comes almost to a standstill and barely turns over the centers. It would thus appear that only a small ratio of expansion in the steam cylinder could be employed; but the difficulty is met by the inertia of the flywheels, and the cylinders of even simple compressors can be proportioned for a steam cutoff at $\frac{3}{8}$ to $\frac{1}{2}$ stroke. In the duplex type, power and resistance are more nearly equalized, the most favorable distribution being attained in cross-compound, stage compressors. The economic advantages of compounding the cylinders are greater than in ordinary steam engines: (1) because the conversion of power from one form to another is always attended by loss, and should be conducted as efficiently as possible; (2) because of the unfavorable load conditions of an ordinary compressor.

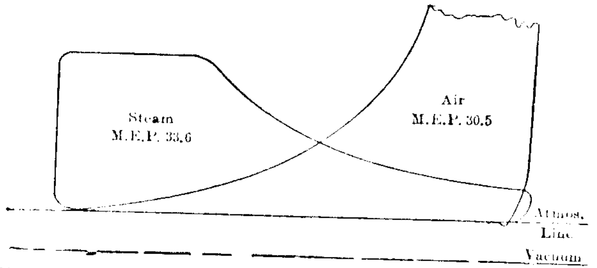

Fig. 1.—Combined Air and Steam Cards, Straight-line Compressor.

Balancing a Compressor's Moving Parts.—In the ordinary designs the inertia of the reciprocating parts causes shock and vibration conditions, that may be dealt with as follows:

I. At each end of the stroke of a horizontal compressor, the maximum inertia corresponds to the centrifugal force of the combined mass of all the reciprocating parts. This is shown in Fig. 2, *a*, the unsymmetrical shape of the shaded area within the curve being due to the varying angularity of the connecting-rod. If an equivalent weight is attached to the crank, opposite the crank-pin, its centrifugal force will act at right angles to that of the compressor's reciprocating parts, as in Fig. 2, *b*. But, though the opposed inertia forces are thus practically neutralized, considerable vibration continues.

II. If, in a belt- or motor-driven, two-stage compressor, one cylinder is placed horizontally and the other vertically, the unbalanced forces are reduced, as indicated by Fig. 2, *c*. In this diagram, from diagrams *a* and *b* are plotted curves *A* and *B*, of which the resultant curve *C* roughly approximates a circle. The fluctuations of the remaining disturbing forces can be read off by means of the accompanying circles, marked 1,000, 2,000 and 3,000 lb.

III. The remaining unbalanced forces can be still further reduced by attaching a counterweight to the crank, as in I, above. This is done in the Sullivan angle-compound compressor (Fig. 33, Chapter III). The result is shown by curves *D* and *E* of Fig. 2, *c*, here, which include between them shaded areas representing the amount of the small residual unbalanced forces. These, due to the varying angular movements of the connecting-rod, can be offset by introducing two additional counterweights, on auxiliary shafts revolving in opposite directions at twice the compressor speed (see Kent's Mechanical Engineers' Handbook, Tenth Edition, p. 1079 and Peele's Mining Engineers' Handbook, Second Edition, pp. 2100-1); but this is a practically unnecessary complication.

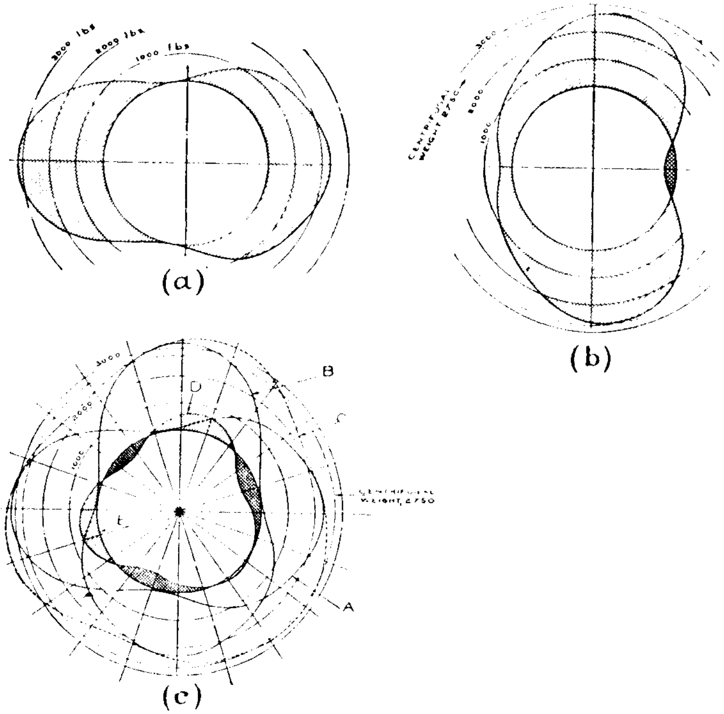

FIG. 2.—Inertia Diagrams of the Reciprocating Parts of a Compressor (Sullivan Machinery Co.)

Among other advantages of balancing a compressor's moving parts are: lower costs of installation and of power, small floor space required, and adaptability to different kinds of drive. Also, as in the "half-duplex," horizontal compressor, previously referred to, the capacity of a single angle-compound compressor can be doubled, if necessary, by adding a second unit, with the motor between the two. This forms the Sullivan Twin Angle-compound (see table of specifications in Chapter III, accompanying Fig. 33).

The Reavell Quadruplex compressor (Figs. 12 and 32, Chapter III) may also be cited here as operating at nearly constant torque; especially when a pair of them are mounted side by side, with a common drive between them. When belt-driven, a practically perfect balance is attainable by casting counterweights on the pulley.

Still other examples of nearly constant torque compressors are those of the Turbo class, and the Hele-Shaw-Beacham rotary compressor (Fig. 51, Chapter III).

Steam Valves of Compressors are in general of the same types as those of ordinary steam engines, and comprise the: plain slide-valve (with or without Meyer cutoff), Corliss valve, balanced piston-valve and drop-valve. (For air valves, see Chapter VII.)

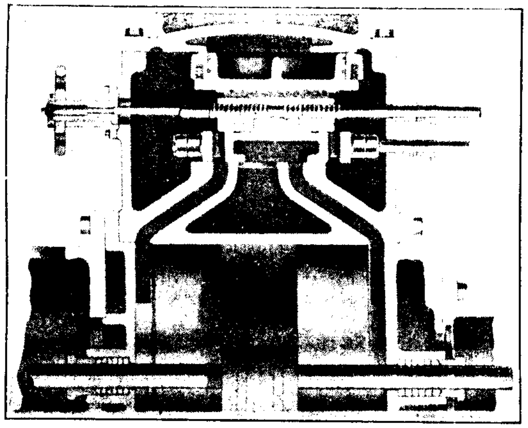

FIG. 3.—Steam Slide-valve, with Meyer Cutoff.

The slide-valve, with adjustable cutoff, is satisfactory for compressors of moderate size, using ordinary steam pressure. They are designed for cutoffs at one-quarter to three-quarters stroke. Some small compressors have balanced slide-valves, without adjustment. The Meyer cutoff-valve rides on top of the main valve (Fig. 3) and, without stopping the compressor, is adjustable by a handwheel for regulating steam admission according to the variations of receiver pressure. For high steam pressures or superheated steam, slide-valves are apt to warp and leak.

The Corliss valve has long been used for large compressors, as it is well adapted to deal with variable load conditions (Figs. 16 and 34, Chapter III). Corliss-type valves were formerly often employed for air cylinders also, but have been almost entirely superseded by different forms of the "thin-plate" valves (Chapter VII).

The balanced piston-valve (Fig. 4; see also Chapter III, Figs. 2, 11 and 17) is widely used for high-pressure steam. The steam enters the middle portion of the main valve and is exhausted past the ends. The accompanying cutoff valves, one for each end of the cylinder, are right- and left-hand threaded on the cutoff valve stem, which telescopes through the main-valve stem. The cutoff-valves are adjustable, as shown. Rotation of their stem in one direction shortens the cutoff; rotation in the opposite direction lengthens it. As there are only two ports, clearance and condensation surface are reduced. The valve covers and packing are under exhaust pressure only, thus decreasing leakage, and subjecting to low temperatures only the walls of the steam chest. For this reason, less lubricant is required.

NOTE. In the case of compound steam cylinders, the low-pressure cylinder commonly has a slide-valve.

The drop-valve is also adapted to compressors working with high steam pressure and superheat (see Chapter III, Fig. 20). It is a double-ported poppet, lifted by a cam and closed by a spring, nearly balanced, and is suitable for fairly high piston speeds. The valves of the high-pressure cylinder admit steam at a fixed point, but an increase of receiver pressure beyond a predetermined point causes an earlier cutoff; the reverse conditions, a later cutoff. The low-pressure admission valves and all exhaust valves are set for constant points of admission and cutoff.

Proportions of Air Cylinders. A short stroke is conducive to economy in compression. With a long stroke the piston travels some distance under an increasing resistance; then, after the discharge valves open, it completes its stroke under a uniform resistance, while adding nothing to the amount of useful work. On the other hand, the loss of volumetric capacity due to piston clearance is less for a long than a short cylinder of the same diameter. In straight-line, single-stage compressors, the usual ratio of stroke to diameter of air cylinder is from 1 : 1 to about 1¼ : 1. In two-stage compressors are found quite wide variations in the cylinder proportions, as indicated by the representative examples from nine different makers shown in the accompanying table.

Cylinder diam., in.		Stroke, in.	Ratios of length of stroke to diameter of cylinder
L.P.	H.P.		
10	6	8	1.25 and 0.75 : 1
11	7	10	1.1 and 0.7 : 1
15	9	10	1.5 and 0.9 : 1
23	14	10	2.3 and 1.4 : 1
18	10	12	1.5 and 0.83 : 1
20	12	14	1.43 and 0.86 : 1
26	15	18	1.44 and 0.83 : 1
32½	19½	23½	1.38 and 0.83 : 1
20	12	30	0.66 and 0.4 : 1
35	22	30	1.17 and 0.73 : 1
38	22	42	0.9 and 0.52 : 1
52	32	60	0.81 and 0.53 : 1

The relative diameters of the air and steam cylinders depend obviously on the steam pressure carried and the air pressure to be produced (see tables of specifications in Chapter III).

Differential Air Cylinders.—Compressors embodying this unusual feature are built by a few makers; for example, by Sir W. H. Bailey & Co., Manchester, England, and Dujardin & Cie., Lille, France. The former is illustrated by a table of specifications and Fig. 17, in Chapter III. The air cylinder is bored of two diameters for the differential piston, two-stage compression being thus effected in one cylinder, with no more moving parts than for single-stage compression.

Compressors Driven by Oil Engines, of both the stationary and portable types, are made by most compressor builders. They use gasoline, fuel oil or low-grade distillate. The stationary type is shown in Chapter III by Figs. 5, 27, and 28, with their accompanying specifications; the portable, by Figs. 39 to 42.

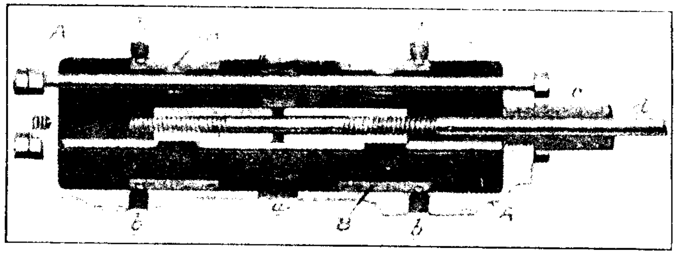

FIG. 4.—Piston-valve (Ingersoll Rand Co.). A, main valve; *BB*, cutoff valves; *a*, steam admission; *b*, ports to cylinder; *c*, main-valve stem; *d*, cutoff-valve stem.

The fuel oil-engine of Class "PO" compressor of the Ingersoll Rand Co. will serve as an example. It is of the single-cylinder, single-acting, four-cycle type, with spring-controlled valves. The oil is injected, by a small cam-driven plunger pump, through two opposed nozzles producing finely divided sprays. Just before the oil is injected, air is drawn into the combustion chamber during the suction stroke, and is compressed during the compression stroke, the mixture of oil and air being ignited by the heat of compression. In the oil-driven compressors of the Chicago Pneumatic Tool Co. (Fig. 5, Chapter III), the power cylinder is of the valveless, two-stroke-cycle, low-compression type. The oil-injection pump is driven by an eccentric on the crank-shaft. The cylinder head, containing the primary combustion chamber, is completely water-jacketed; the cylinder jackets cover only the portion of the cylinder in which combustion takes place, to minimize stresses due to unequal heating. For starting fuel oil-engines, high-pressure air is necessary, which is supplied by a small auxiliary, single-acting, gasoline-driven compressor.

Fuel-oil-driven compressors are economical and advantageous for many kinds of service. Gasoline engines are obviously especially convenient for driving portable compressors, except for underground use in mines; in which case, due to questions of ventilation, electric drive is necessary.

Lubrication of Compressors.—The main bearings of most modern American compressors, together with some of European make, are self-oiling, by the "flood and splash" system (for examples, see Figs. 1, 3, 5, 9, 23, 25 and other illustrations in Chapter III). For this purpose, the main bearing, crank-pin and crosshead are entirely inclosed. The crank-pit contains the oil supply. The oil is picked up by the edge of the crank disk, and taken off at the top by a scraper; part goes to a distributing tank, from which small pipes lead to the crosshead and guides. The main bearing and crank-pin are oiled direct from the scraper by a projecting trough on each side. For the bearing the oil is generally led to a channel in the cap, and thence through a series of holes drilled in the bearing liner. Some of this oil goes to the collar of the eccentric hub, from which it is carried to the face of the eccentric through two holes. As the eccentric is inclosed, the surplus oil returns to the crank-pit. Thus the oil-feed is proportioned to the speed of the compressor, ceasing entirely when the compressor is stopped, and a material saving results.

Comparison of Modes of Drive.—Steam-driven compressors are applicable to a wide range of service, but do not operate with a high degree of economy unless the steam end has compound cylinders.

Electric drive is in general the most economical, if electric current is already provided for other purposes, or purchased power is available. As shown by Figs. 6, 9, 19, and 26, Chapter III, the motor may be belted or direct-connected to the compressor. When direct-connected the armature is mounted on the crank-shaft, making a self-contained unit with a simple foundation. The armature, being made of large diameter to produce a proper relation between peripheral and rotative speed, serves also as a flywheel. Induction motors are advantageous in operating economically under wide variations of load; but when direct-current motors can be used, they are handier and more easily controlled. Synchronous (constant speed) motors are also applicable.

With belt drive, the short belt and idler pulley are adopted if floor space is a consideration (Fig. 9, Chapter III), but, mechanically, the long belt is preferable when there is enough room.

Tooth gearing has been occasionally used instead of a short-belt drive from a motor, for small compressors erected underground in mines, where space is restricted. Belting deteriorates rapidly in warm, humid mine air; it must always be protected from dripping water. Gearing is objectionable, because of its noise and liability to breakage. Noise can be reduced by a rawhide pinion. "Silent-chain" drive has been employed in rare instances; now considered obsolete.

CHAPTER V

THE AIR CYLINDERS OF COMPRESSORS

For many years following the invention of compressors, the mode of dealing with the heat generated in the air cylinder led to a basic distinction. In *wet compressors* cooling water was introduced into the cylinder in bulk and also in the form of spray; or it was injected only as sprays or jets. In *dry compressors* no water enters the cylinder, except that which is carried as atmospheric moisture in the air itself; all cooling during compression, aside from that due to radiation, is effected by water-jacketing the cylinder.

All of the earlier compressors, from 1861 to about 1875, were of the wet type. They included the historically interesting Colladon, Dubois-François and Humboldt compressors, used in driving the Mont Cenis and other Alpine tunnels; and the Burleigh, which was one of the group of pioneer American compressors that may be said to have been invented and developed specifically for use at the four and one-half mile Hoosac tunnel, in Massachusetts, the first long tunnel in the United States (see Chapter I of this book; also Chapter V of "Tunneling, Explosive Compounds and Rock Drills," by Drinker). Until quite recently, a few of the Humboldt wet compressors were still in operation in Europe.

The old wet compressors were followed, after 1875, by the Rand, Clayton, National, Norwalk, Ingersoll and others. In several of these the idea of dry compression took shape, and their designs embodied many of the outstanding features of present-day practice.

Wet *versus* Dry Compression.—Wet compressors, which necessarily run at a slow speed, were undoubtedly efficient from the thermodynamic standpoint, as injected water cools the air more thoroughly than is possible by water jackets.

In well-designed two-stage, dry compressors, working at 6 atmospheres (73.5 lb. gage) delivery pressure, the loss of work due to heating of the air is reduced approximately one-half (from 24 per cent to about 12.5 per cent; in ordinary single-stage, dry compressors, the loss would be, say 16 per cent. By spray injection, well intermixed with the air, the heat loss for the same delivery pressure may be as low as 4.5 per cent; occasionally only 2 to 2.5 per cent in large, slow-speed, wet compressors.

In purely adiabatic compression (Chapter II) the theoretical loss due to the conversion of work into heat is as follows:

For compression to	Equivalent pounds gage pressure	Loss, in per cent
2 atmospheres	14.7	9.2
3 atmospheres	29.4	15.0
4 atmospheres	44.1	19.6
5 atmospheres	58.8	21.3
6 atmospheres	73.5	24.0
7 atmospheres	88.2	26.0
8 atmospheres	102.9	27.4

But there are other considerations affecting the work of compressors.

First, as to the effect of injected water upon the compressed air and the machines using it. The amount of heat absorbed during wet compression is proportional to the difference of temperature between the intake air and the injected water, and to the time of contact between the air and water. This difference of temperature, usually *nil* at the beginning of the stroke, reaches its maximum at the end. Hence: (1) to attain a fair approach to isothermal compression the piston speed must be very slow; (2) during the first part of the stroke but little heat is abstracted, and it is only after discharge begins that the cooling effect is maximum. Therefore, since much of the total heat must be given up after compression is completed, the lower final temperature of wet compression is in a measure deceptive. The heat remaining at discharge augments the moisture-carrying capacity of the air, and considerable moisture passes into the transmission pipe, deposits in them as the air cools, and in cold weather may freeze and reduce the effective diameter of the pipe. Furthermore, the moisture remaining in the air may cause troublesome accumulations of ice in the exhaust passages of drills or other engines using the air (Chapter XVI). In the dry compressor, although the moisture always present in atmospheric air makes its appearance as frost at the exhaust of the air machine, there is rarely enough to cause serious trouble.* Delivery of warm air by a dry compressor is less disadvantageous than warm air from a wet compressor.

Second, as to the effect of injected water upon the working of the compressor. Water in the air cylinder is mechanically objectionable, because it makes lubrication difficult, causes rust, and increases wear. Injected water must be pure and free from grit. Water that is harmless for use in jackets may be injurious to valves, cylinder and piston.

Conclusions.—A large, slow-speed wet compressor, with abundant cold injection water, will give a good indicator card, but its first cost is greater per unit of output than that of a dry compressor. The friction loss is higher in the wet than in the dry compressor. Wet compressors are now entirely obsolete. For further information regarding their design and operation, see Chapter IV of the previous editions of this book.

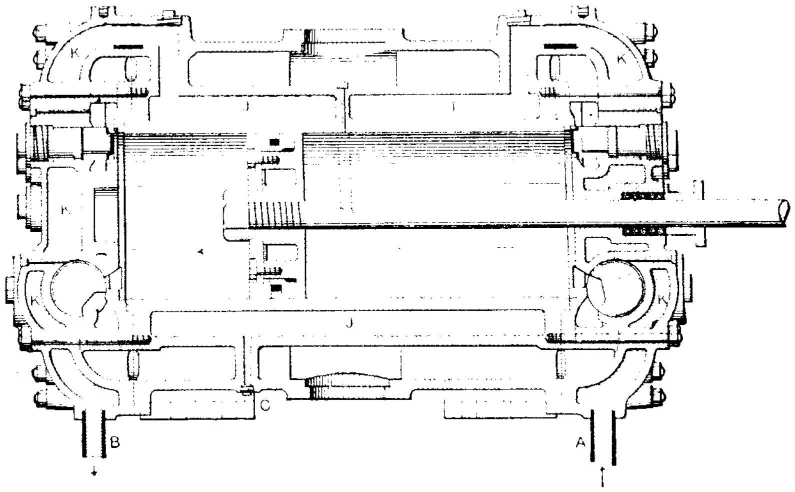

FIG. 1. A Well-jacketed Air Cylinder. (The valves shown are now obsolete.)

All modern compressors are of the dry type. Details of representative designs of air cylinders of different makers follow. Still other examples are shown in the illustrations in Chapter III.

Water-jackets.—These comprise annular, peripheral jackets, surrounding the barrel of the cylinder, and end jackets, on the cylinder heads. Their design should be based upon the thermal behavior of the air while being compressed (Chapter II).

The temperature of the air in the cylinder is highest when the pressure reaches its maximum; that is, when the piston is approaching the end of its stroke during the discharge period. Therefore the cylinder heads should be especially well jacketed, with active circulation of the cooling water. In many of the older compressors, the valves occupied so large a part of the area of the cylinder head that little space was left for the end jackets. Fig. 1 shows an exceptional design (of an old style cylinder, with the valves in the heads), in which the air passages and the valves themselves are quite fully surrounded by the end jackets. In most air cylinders of the present day, the area of the end jackets is readily made larger, and their effect is greater, than formerly was generally thought possible; because the valves of nearly all compressors are now placed radially around the periphery of the cylinder, close to the ends, instead of in the cylinder heads.

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* The quantity of moisture in the atmosphere, or its humidity, varies with the climate, altitude above sea-level, and the season of the year. It is usually greatest near any large body of water. What is commonly called dry atmospheric air contains from 40 to 50 per cent of the quantity of moisture necessary to saturate it. The degree of saturation in summer often reaches 90 per cent or more. For a fuller discussion of this subject, see Peele's "Mining Engineers' Handbook," Second Edition, Sec. 23, Art. 1.

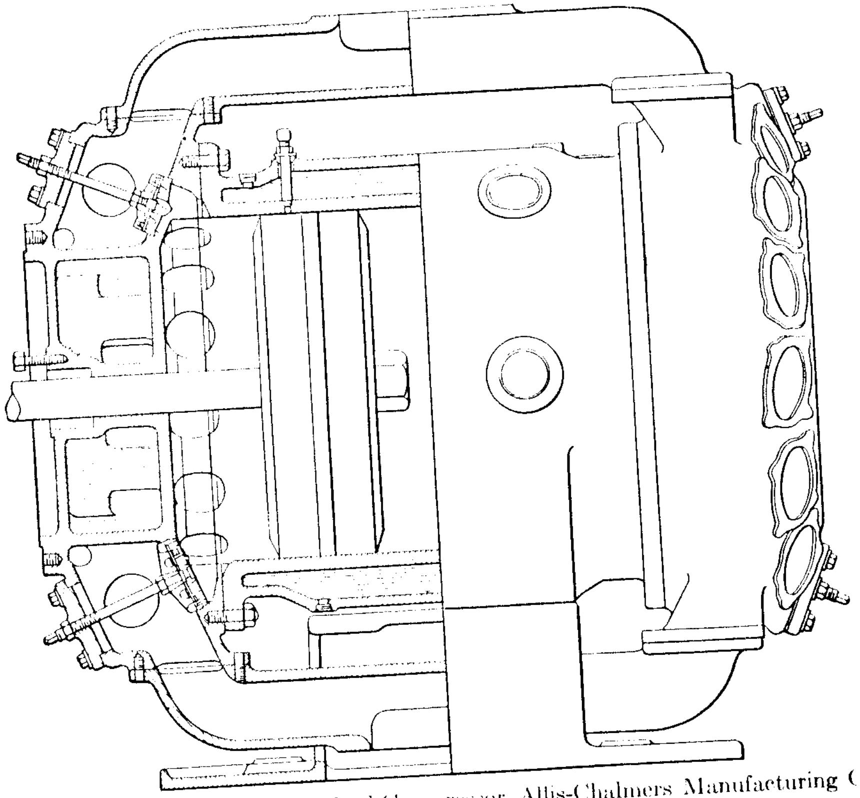

Fig. 2.—Air Cylinder for Standard Compressor, Allis-Chalmers Manufacturing Co.

The peripheral jackets are less effective directly; they are rapidly passed over and covered by the piston as it advances in its stroke, and therefore only a small part of the jacket area can act in cooling any given cylinderful of air, when at maximum pressure and temperature. Moreover, only a part of the air in the cylinder comes into immediate contact with the jacketed surfaces. Figs. 2 to 5 show typical examples of air-cylinder design.

But this direct action of the water-jackets upon the air while under compression is supplemented by their important function of keeping

down the temperature of the cylinder as a whole. The jacket. [illegible]ol the entire cylinder walls, thus preventing possible burning or decomposition of the oil and aiding efficient lubrication of the piston and inner surface of the cylinder (see below). With this end in view, the cylinder walls should be as thin as is consistent with safety, allowance being generally made for one re-boring of the cylinder. It has even been proposed to make the inner shell of the cylinder of hard brass, which by its high conductivity would assist in carrying off the heat.

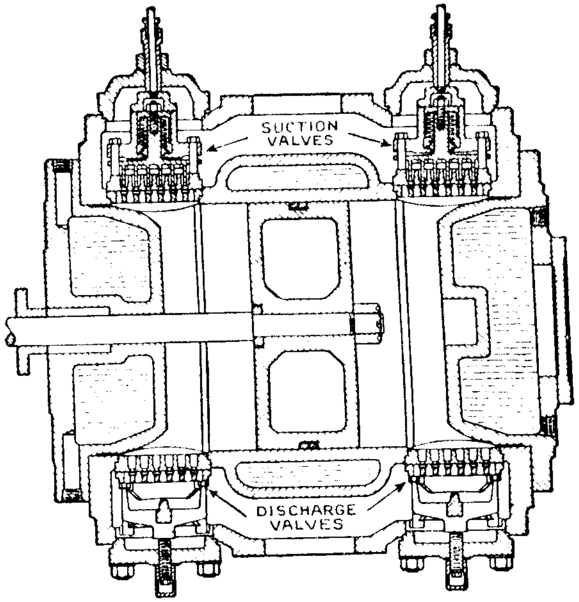

Fig. 3.—Air Cylinder of the Worthington Compressor.

To insure active circulation of the cooling water, the jacket spaces should be subdivided, as indicated in Fig. 1. The water, entering at *A*, passes thence through the peripheral and end jackets, *JJ* and *KK*, and is discharged at *B*. At *C* is a drain pipe, for blowing out sediment from time to time. Hand holes are also provided for removing hard, scaly accumulations. In some designs, the peripheral jacket is divided by partitions, so that the water passes first around about one-fifth of the length of the cylinder nearest the ends, and then around the middle portion, from which it is finally discharged. In still other designs, the water first enters the end jackets and thence passes successively through separate compartments of the peripheral jacket. Fig. 6 shows a jacket having eight radial partitions, extending alternately from each end of the cylinder nearly to the opposite end. The water is compelled to flow back and forth through the longitudinal spaces thus formed. The jacket water is generally supplied by gravity from an elevated tank; rarely, by a small pump operated from the crosshead of the compressor.

Water Supply for Cylinder Jackets (Ingersoll-Rand Co.)

Gallons per 100 cu.ft. of free air

	Temperature of water	60° F.	70° F.	80° F.	90° F.
Single-stage compression	For air pressure of, lb.:				
	40	0.5	0.6	0.7	0.9
	60	0.6	0.7	0.8	1.0
	80	0.7	0.8	0.9	1.1
	100	0.8	0.9	1.0	1.2
Two-stage compression	Jackets of both cylinders, pressure of, lb.:				
	80–100	0.85	1.0	1.2	1.4
	Jackets and intercooler in series, pressure of, lb.:				
	80–100	2.9	3.4	4.0	4.5

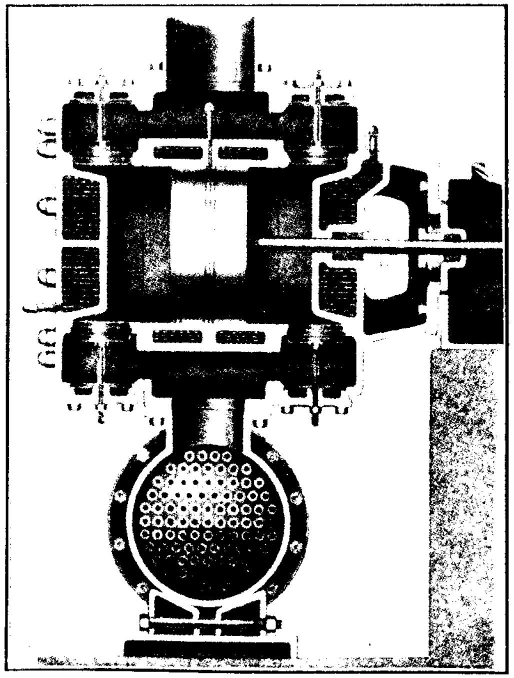

Fig. 4.—Ingersoll-Rand "Imperial" Air Cylinder, for Types XCB and XRE Compressors.

The useful effect of water-jackets also depends largely upon the running speed of the compressor. In the best single-stage compression, say to 70 or 75 lb. and at not over 300 ft. piston speed, probably not more than one half of the total possible cooling can be effected; that is, the exponent n, in the general equation for adiabatic compression (Chapter II), would be equal to, say, 1.25. Even when a compressor is running at normal speed, the cylinder, discharge pipe, and the receiver, are usually quite hot; often too hot to be touched with the hand. With well-jacketed cylinders, and compressing to only 45 lb., the temperature of the air at delivery has reached 280° F. The heat of compression in dry compressors probably ranges from 200° to 400° F. for the ordinary pressures of 75 to 100 lb. used in mining, though it does not often exceed 350°. For proper lubrication of the cylinder, the temperature should be kept below this point (see Chapter XIII). At a mine in Montana, the writer has observed the thin wrought-iron delivery pipe of an over-

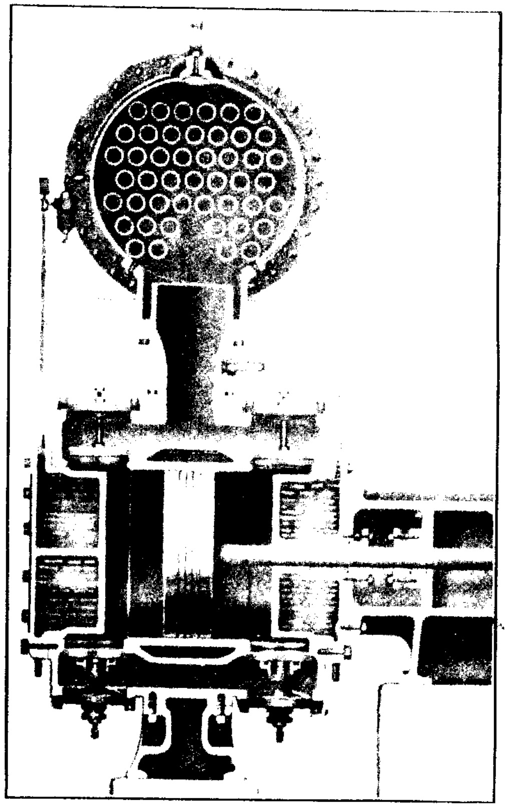

Fig. 5.—Air Cylinder of Class O-CB Compressor, Chicago Pneumatic Tool Co.

driven compressor to be red-hot for a distance of nearly 6 in. from the cylinder shell. Driving compressors at too high a speed not only causes poor thermodynamic results, but the excessive heating of the cylinder may decompose the lubricating oil. The gases thus formed will produce a foul, noxious exhaust from rock-drills, which may become troublesome or even dangerous in poorly ventilated mine workings (Chapter XIII).

Air Valves.—The design and operation of the inlet and discharge valves are discussed in detail in Chapter VII.

Cylinder or Piston Clearance.—Toward the end of the stroke, the compressed air in front of the piston begins to pass out through the delivery valves when its tension exceeds that of the air in the discharge pipe to the receiver. But, on completing the stroke, a certain quantity of hot compressed air remains in the clearance space between the face of the piston and the cylinder head. On the back stroke this clearance air expands behind the piston, and no fresh air can enter through the inlet valves until the cylinder pressure falls below atmospheric pressure. Hence: (1) the volumetric capacity per stroke, in terms of cu.ft. of free air, is always less than the volume air, is always less than the volume swept through by the piston; (2) the linear clearance must be made as small as possible. It is usually about $\frac{1}{16}$ or $\frac{1}{20}$ of an inch. NOTE. In the old wet compressors the clearance space was filled with water, and did not affect the volumetric capacity.

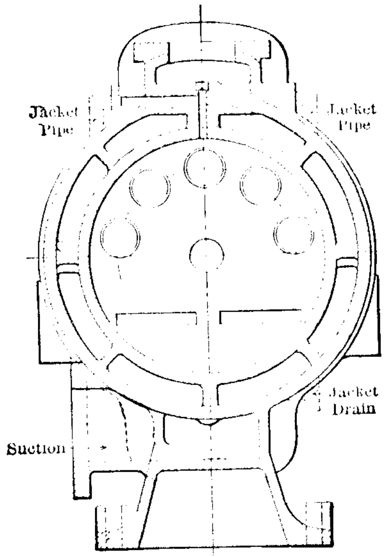

FIG. 6.—An Air Cylinder with Radial Partitions in the Peripheral Water Jackets.

Fig. 7 shows the effect of clearance, the thermodynamics of which are exhibited by formulas (21) to (36) of Chapter II. Before the inlet valves can open, the piston travels from c to b, the corresponding cylinder volume representing the loss of volumetric capacity. The actual effect of clearance on the volumetric efficiency of a compressor depends on the delivery pressure. The higher this pressure, the greater is the distance cb. It should be noted that: (a) the loss of volumetric capacity does not involve a corresponding loss of useful work (see text under equation 24, Chapter II); (b) the compressed air remaining in the clearance space helps to overcome the inertia of the moving parts at the beginning of the return stroke; and (c) in re-expanding behind the retreating piston, the clearance air cools rapidly and does not materially raise the temperature of the incoming atmospheric air. The effect of clearance in reducing the volumetric capacity of single-stage compressors is also shown by Fig. 8. In long-stroke compressors, the clearance volume is less than in short-stroke compressors.

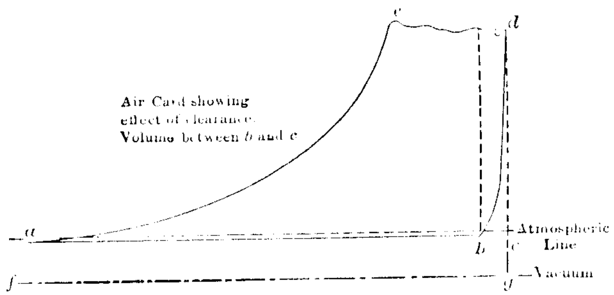

FIG. 7.

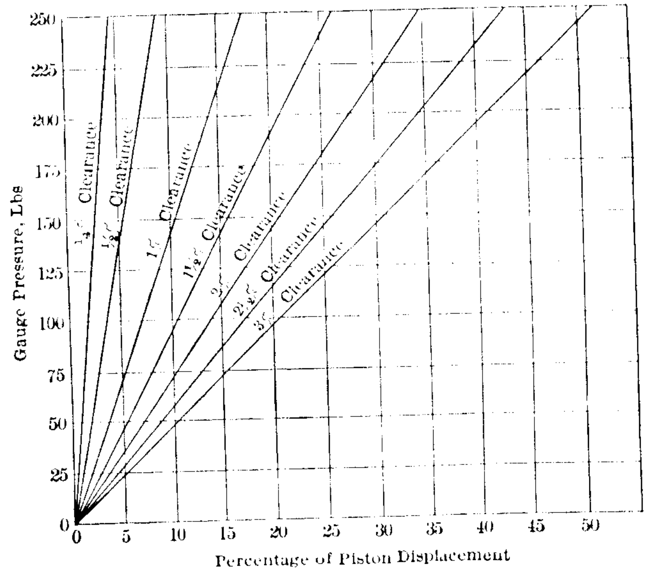

FIG. 8.—Effect of Piston Clearance.

Examples of total linear clearances in single-stage compressors:

Stroke, in.	Clearance	Stroke, in.	Clearance
14	0.0190	36	0.0112
21	0.0176	48	0.0093
24	0.0126		

thus ranging from about 2 per cent down to 1 per cent.

In two-stage compressors, the total clearances are larger, as shown by these examples:

	Per cent		Per cent
28 × 24 in.	2.15	18 × 14 in.	2.00
23 × 20 in.	1.70	17 × 14 in.	2.19

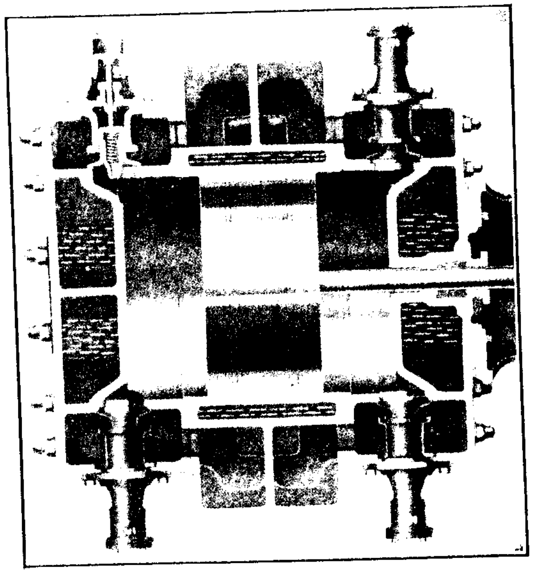

Fig. 9.—Cylinder with Valve Clearance Pockets. Ingersoll-Rand Compressor, Type PRE.

To sum up, on the basis of the figures cited, the best range of percentage of clearance volume to total cylinder volume, as read on the lower horizontal line of Fig. 8, is from say 3 to 15 per cent.

The larger percentages are found in cylinders having their valves arranged peripherally (Figs. 3, 4, 5 and 9). In these, the volume of the valve pockets increases the clearance. Incidentally, by thus placing the valves (instead of in the cylinder heads, as formerly), space is afforded for larger and more efficient water jackets on the cylinder heads. Although the valve pockets are partly covered by the piston at the end of its stroke, the discharge flow of the air is but little obstructed.

To minimize clearance, the piston must be accurately adjusted to its length of stroke. When the linear clearance is very small, care must be taken, when fitting the crank-pin or crosshead-pin with new brass bushings, to avoid a change in length of stroke that might cause the piston to strike the cylinder head.

If the piston is bolted to the piston-rod, with a projecting nut, a recess is cast in the cylinder head to receive the nut at the end of the stroke (Fig. 3). Usually, however, the piston is shrunk on its rod, so that a recess is unnecessary.

To avoid the disadvantages of clearance, some old-style compressors had grooves cast in the inner surface of the cylinder near the ends. When these were uncovered by the piston at the end of its stroke, the high-pressure clearance air was released to the other side of the piston. An objection to such a device is that the sudden removal of the heavy pressure on the piston causes hurtful shocks.

When a compressor is operated by a synchronous motor, the starting characteristics of the motor require that the air cylinders must be provided with "relief valves." These are opened by hand during the starting period, thus temporarily by-passing the discharge air to the atmosphere.

CHAPTER VI

OPERATION OF STAGE COMPRESSORS

STAGE compressors divide the work of compression between two or more cylinders. In two-stage compressors air at atmospheric pressure enters the intake or low-pressure cylinder, is there compressed to a certain point, and is then forced into the high-pressure cylinder, where it is brought up to the required tension. The cylinders are proportioned to divide the total work equally between them. This secures equalization of resistances, and promotes the efficiency of the cooling apparatus.

The theory is given in Chapter II, and examples of standard designs in Chapter III. Though some of the heat of compression is dissipated by radiation, the working temperature in the cylinder of a single-stage compressor is too high to be dealt with effectually by the water-jackets, because in a single cylinder the area to which cooling can be applied is small relatively to the volume of air, and the compression period too short. Even at moderate piston speeds (350–400 ft. per min.), the cooling is so imperfect that the compression line of the air card approaches closely to the adiabatic line. Moreover, the high temperature makes it difficult to lubricate the air cylinder properly. In compressing to 90 lb. gage in one cylinder, the theoretical final temperature is 459° F. (See first part of Chapter XIII.)

Formerly, stage compression was employed for high pressures only, as for pneumatic locomotives, riveting machines, compression of gases, etc. To produce very high pressures (500–1,000 lb. or more) three- and four-stage compressors are necessary.

It is now recognized that two-stage compression is advantageous even for pressures of 75–90 lb., as commonly used for machine drills. The cooling taking place during compression is more thorough, because, for a given final pressure, the heat generated in each cylinder is less than one-half of that which would be produced in one cylinder, and the total water-jacketed area is larger.

Further cooling is effected by the "intercooler" (Fig. 2), placed between the cylinders. It is an intermediate chamber, through which the air from the intake cylinder passes on its way to the high-pressure cylinder. In it the temperature of the air is reduced, so that the air entering the high-pressure cylinder is considerably cooler than at discharge from the low-pressure cylinder. The total reduction of temperature depends upon the volume of air under compression, temperature of the water supply, area of cooling surfaces and length of time the air is in contact with these surfaces; the last named factor depending in turn upon the piston speed and the design of the compressor.

Range of Work for which stage compression is applicable:

1. Stage compression is of doubtful utility for gage pressures of much less than 75 lb., because of the small saving as compared with the greater cost of the more complicated mechanism. It is generally applicable for pressures higher than say 75 lb.

2. Stage compression is especially useful for large plants, in which the percentage of saving represents an amount sufficient to warrant the greater first cost.

3. The higher thermodynamic efficiency of stage compression is partly offset, and in poorly designed compressors may be entirely neutralized, by the increased frictional losses due to the use of several cylinders. A well-designed, economical steam end should be used, together with efficient cooling for the air end; otherwise multiple-stage compression may cost more per cubic foot of air delivered than compression in a good, single-stage compressor.

All ordinary stage compressors are double-acting; that is, on each forward and back stroke air is taken into the cylinders on one side of the piston, while compression and delivery take place on the other side. In the single-acting form the resistances in the cylinders are not as well equalized throughout the stroke. This form is always employed for the high-pressure cylinders of locomotive chargers (Chapter XXIV) and of gas compressors, because of the difficulty of maintaining air valves for very high pressures, to give the water jackets more time to act, and to simplify the heavy castings required.

Double-acting Stage Compressors.—In these the cycle of operations during each forward and back stroke is the same, and hence the distribution of resistances is more uniform.

Fig. 1 shows the air end of a two-stage compressor, with tandem cylinders. When the pistons reach the end of their forward stroke, the conditions in the two cylinders are as follows: The intake cylinder (*B*) is full of air, practically at atmospheric pressure, while the high-pressure cylinder (*F*), with the intercooler (*E*) and connecting passages, are occupied by air just delivered from the intake cylinder. On the reverse stroke the free air in front of the intake piston is compressed to somewhat less than one-half the final pressure, and delivered through the intercooler into the high-pressure cylinder, while the air already occupying the latter is brought to the final pressure and discharged.

Proportions of Air Cylinders.—In standard two-stage compressors, the typical volumetric capacities of the low- and high-pressure cylinders are to each other in the ratio of about 10 to 4, to proportion them so that their ratios of compression are nearly equal. The distribution of work and the heat generated in the cylinders are then approximately equalized and most effectually dealt with by the intercooler, which is of primary importance in stage compression, and is now made of much greater capacity than formerly. The hot air from the intake cylinder is kept longer in contact with the cooling surfaces, because of its reduced speed of flow through the large cross-sectional area of the intercooler, and it enters the high-pressure cylinder at a correspondingly lower temperature. The connections between cylinders and intercooler should be of as small

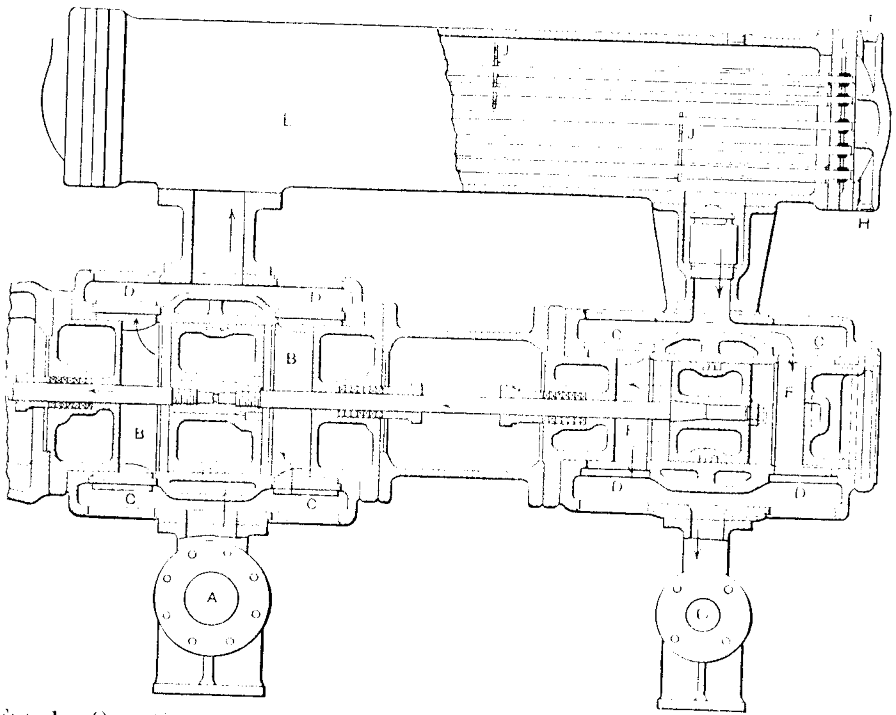

Fig. 1.—Operation of Tandem, Two-stage Compressor. (Norwalk Co.) When the pistons move in direction of the arrows on the piston-rod, the air circulates as shown by arrows in the cylinders. *A*, air inlet; *B*, intake cylinder; *C*, *C*, inlet valves; *D*, *D*, discharge valves; *E*, intercooler; *F*, high-pressure cylinder; *G*, discharge air pipe; *H*, water inlet to intercooler; *I*, water outlet; *J*, air baffle plates. The cylinders in the cut are 12 in. and 7½ in. in diam. by 8-stroke.

volume as is consistent with reasonable frictional resistance to the flow of air; because the air occupying these passages at any given time cools slightly by radiation only.

If the net volume of the intake cylinder is 10, the volume of its connection with the intercooler should be about 1.5, of the intercooler 10, of the connection to the high-pressure cylinder 1.5, and of the high-pressure cylinder 4. With these proportionate volumes, the cycle of operations during a single stroke is as follows. Assume the stroke to be from right to left, as indicated in Fig. 1. By the previous stroke the intercooler and its connections to the cylinders, representing a volume = 1.5 + 10 + 1.5, were filled with air at, say 35 lb., for a final pressure of 90 lb. This body of air, then shut off from both cylinders by their valves, loses part of its heat and pressure in the intercooler. During the first part of the next (left-hand) stroke, the intake piston acts only on the cylinderful of free air just taken in (volume = 10). While this is being compressed, the advance of the high-pressure piston causes the air in the intercooler and its connections to flow into the high-pressure cylinder. Beyond mid-stroke the pressure in front of the intake piston rises slightly higher than that in the intercooler and its delivery valves open, so that this piston acts upon the entire body of air: volume = $\frac{10}{2} + 1.5 + 10 + 1.5 + \frac{4}{2} = 20$. Then, until the end of the stroke, both cylinders are in communication through the intercooler, *i.e.*, from the left-hand end of the intake to the right-hand end of the high-pressure cylinder, and an approximate equalization of pressure is established throughout.

Until the left-hand intake delivery valves open, the air in the intercooler is isolated from the intake cylinder, in which compression has progressed without other cooling than that of the cylinder jacket. But when the warm air begins to pass from the intercooler into the high-pressure cylinder, the influence of the intercooler is exerted upon a new body of air. At the end of the left-hand stroke the closing of the delivery valves again shuts off the air in the intercooler from both cylinders. The high-pressure cylinder, on the right-hand side of the piston, is then occupied by a body of air the temperature and pressure of which have been reduced by the combined effect of the intercooler and both water-jackets to a point below that due to the working pressure of the intake cylinder.

In the latter part of the left-hand stroke, when the intake delivery valves have opened and the piston of this cylinder is acting on the volume 20, as stated above, part of this air (volume = $\frac{2 + 1.5}{20}$ = 17.5 per cent of the total) has passed beyond the influence of the intercooler, and another part (volume $= \frac{5 + 1.5}{20} = 32.5$ per cent) has not yet reached it. At the end of the left-hand stroke the volume of compressed air in the intake cylinder = 0 (neglecting clearance), in the intercooler and its connections 1.5 + 10 + 1.5 = 13, and in the high-pressure cylinder 4, a total of 17, of which 1.5 has not reached the intercooler, but has been affected only by the water-jacket of the intake cylinder. This analysis emphasizes the importance of cylinder jackets of large area, with ample circulation of cold water, a large intercooler, and small connecting passages.

The foregoing details apply precisely only to compressors in which the cylinders are tandem. In duplex stage-compressors, the cycle of operations is somewhat different because the pistons, instead of moving together in the same direction, work with their cranks 90° apart.

Though the total work should be equally divided between the air cylinders, still, due to the frequent variations in receiver pressure, upon which depends the exact terminal pressure in the high-pressure cylinder, only an approximate equalization is attainable. On the basis of some terminal pressure taken as normal, such diameters are assigned to the cylinders as will make their compression ratios nearly equal. Take, for example a pair of cylinders, 15 in. and 24 in. in diameter, to compress to 85 lb. gage. Assuming that the air between the stages is cooled to the intake temperature, the absolute intake pressures of the cylinders will be inversely proportional to the squares of their diameters, or: $15^2 : 24^2 :: 14.7 : 37.64$. The ratio of compression in the intake cylinder is then $\frac{14.7}{37.64} = 0.3905$; in the high-pressure cylinder, $\frac{37.64}{85 + 14.7} = 0.3775$. This is as close to perfect equalization as is necessary.

The gage pressure in the intercooler is equal to the absolute intake pressure plus 10 per cent, minus the atmospheric pressure, multiplied by the ratio of the cylinder areas. The added 10 per cent allows for the fact that the air in the intercooler is not completely cooled to the temperature of the low-pressure intake.

The Intercooler (Fig. 2) in its usual form is a long cylindrical chamber, containing parallel, thin brass or wrought-iron tubes, through which cold water is circulated. The air passes through the spaces between the tubes. The intercooler is generally placed horizontally either above or below the cylinders and close to them so that the connecting passages may be short and of small volume, because, as already stated, the air in these passages at any given time is denied the cooling effect both of the cylinder jackets and of the intercooler. Vertical intercoolers are also sometimes used. For "receiver-aftercoolers" and their

important functions, see Chapter IX. The intercooler tubes must be close enough together to split up thoroughly the body of air traversing the intermediate spaces and so secure the maximum cooling effect. The general flow of the water should be opposite to that of the air; the hottest air then comes into contact with the tubes containing the warmest water and *vice versa*. Obviously, the water should be as cold as possible. With perfect intercooling, the temperature of the air, on passing from the intercooler to the high-pressure cylinder, would be reduced to the normal; in ordinary practice, the air is cooled to within say 15° or 18° of the temperature of the water used. The effect upon the compression curve of this drop in temperature is shown by Fig. 3; the high-pressure compression curve should begin close to the isothermal line. Every 10 per cent decrease in the temperature of the air delivered to the high-pressure cylinder decreases by about 1 per cent the power required for compression.

Cooling Water Required for Two-stage Compression

Gallons per 100 cu.ft. free air compressed to 80–100 lb.

Temperature of water supply	60°	70°	80°	90°
Intercooler and jackets in series	2.9	3.1	4.0	4.5
Intercooler only	2.5	3.0	3.5	4.0
Cylinder jackets only	0.85	1.0	1.2	1.4

Brass intercooler tubes are preferable to iron because of their higher conductivity; but iron tubes cost less, and being rougher present a larger cooling surface to the air. They should be as thin as is consistent with the necessary strength. The tubes are expanded into tube-plates at each end, and by baffle-plates the air is compelled to pass through the entire volume of the intercooler. One tube-plate is attached to the shell, the other being free to move as the tubes expand or contract. The end water-heads are so divided that the water circulates actively back and forth several times, before final discharge. In Fig. 2 the air from the low-pressure cylinder entering at *A* passes alternately above and below the successive baffle-plates to the connection at *B* with the high-pressure cylinder. To compensate the decrease in air volume due to cooling, the baffles are spaced closer toward the high-pressure cylinder. Fig. 33, Chapter III shows a "3-pass" intercooler of large volume in proportion to the size of the cylinders. The pressure drop allowed in a well-designed intercooler corresponds to $3\frac{1}{2}$ or 4 in. of water column (= 0.126–0.145 lb. per sq.in.). The inclined baffles in the connections to the cylinders (Fig. 2) form traps for collecting moisture deposited due to cooling, the water being drawn off by drain cocks.

For the high-pressure air between the third and fourth stages of multi-stage compressors, the Norwalk Co. uses an "open-tank" intercooler, as shown in Fig. 30, Chapter III. The tank contains coils of pipe, surrounded by circulating water.

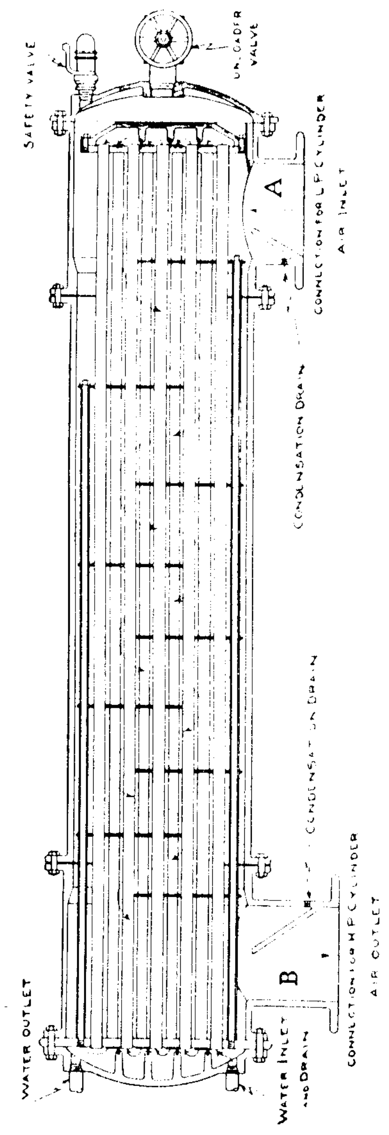

Fig. 2.—Ingersoll-Rand Intercooler.

Comparison of Work Done by Single- and Two-stage Compressors. Frictional losses are omitted, and no account is taken of the cooling due to the cylinder jackets (Chapter V).

1. A single-stage compressor, producing a gage pressure of 70 lb. at sea-level, with 24-in. cylinder and piston speed of 400 ft. per min., will have a capacity of 1,256 cu.ft. per min., in terms of free air at normal temperature. In adiabatic compression, the mean cylinder pressure will be 33.83 lb. and the hp. 184.38.

2. For doing the same work in a two-stage compressor, having an intercooler capable of reducing to the normal the temperature of the air between the cylinders, assume that the intake cylinder has also a diameter of 24 in., and that the pressure produced in it is 35 lb. The mean pressure corresponding to this terminal pressure is 25.6 lb., and the hp., is 118.19. The diameter of the high-pressure cylinder, under the assumed conditions, is found by making the piston area inversely proportional to the increase in absolute pressure of the air delivered to it by the intake cylinder, i.e., in the ratio of 14.7 : 35 + 14.7 = 1 : 3.38. This gives an area of 135 sq.in., or 13 in. diameter. Compressing in this cylinder from 35 to 70 lb. gage, the mean effective pressure will be 28.74 lb., and the **hp.** is 46; or a total for both cylinders of 118.19 + 46 = 164.19 hp.

This shows a saving by two-stage compression of: 184.38 − 164.19 = 20.19 hp., or about 11 per cent. The theoretically perfect cooling between the cylinders here assumed is unattainable, and the frictional loss in the stage compressor would probably be a little greater than in the single-cylinder machine; so that the net gain due to intercooling may in this case be taken at, say, 7–8 per cent. The saving is increased in dealing with higher pressures. (For "Stage Compression at High Altitudes," see Chapter XII.)

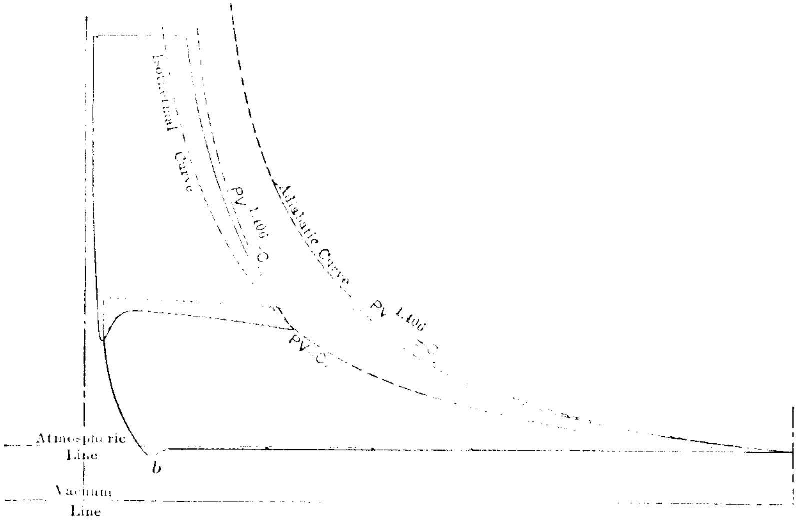

Fig. 3.—Combined Air Cards of Two-stage Compressor (F. A. Halsey). (Minor irregularities in the lines are eliminated.)

Initial cooling of the intake air may be adopted with advantage, especially in hot climates. This is done by "antecoolers," either horizontal or vertical, similar to the "aftercoolers" illustrated in Chapter IX.

Air Card of Two-stage Compressors.—The compression curves of a two-stage compressor are shown in Fig. 3, the adiabatic and isothermal curves being also laid down. These cards are from a pair of $7\frac{1}{2}$ and 14 × 16-in. cylinders, compressing to 110 lb. gage, at 135 rev. per min., or 360 ft. piston speed. Temperature of cooling water: initial, 55°; at discharge from jackets and intercooler, 62° F. Several other points are to be noted.

(1) The overlapping of the high- and low-pressure cards indicates a loss, the work represented by the area of overlap being in reality *work*

done twice. It is due to the drop in pressure between the cylinders, caused by valve resistance and the friction in air passages and intercooler. This pressure drop is minimized by making the valves, ports, and connecting passages of ample size.

(2) As with single-cylinder compressors, the compression line of each cylinder of most stage compressors departs but little from the adiabatic curve. The intercooler is the chief element in economical working. By its abstraction of heat the volume of air entering the high-pressure cylinder is reduced, so that $PV^{n=1.4} =$ constant becomes approximately $PV =$ constant, on beginning the second stage. But the compression line again rises rapidly from this point and continues not far below the adiabatic line. Indicator cards that do not show approximately this relation are always open to suspicion. A leaky piston, for example, will lower the compression curve and make it appear that better work is being done than is really the case.

In constructing and reading a combined indicator card from a stage compressor, the adiabatic line applying to the compression in the second cylinder should be represented in its proper place (see Fig. 3). The complete graphic relation between the several heat curves is thus set forth.

(3) It is an advantage of stage compression that there is practically but one clearance space—that in the intake cylinder, and, as the air in this cylinder is at a low pressure, the resulting reduction in volumetric capacity is moderate, because the loss due to clearance is proportionately less for low than for high pressures. It is important to note that the piston clearance in the high-pressure cylinder can not affect the volume of air delivered, because all the air discharged from the intake cylinder goes to the high-pressure cylinder and, barring leakage, must pass through it.

The heating of the cylinder walls and pistons reduces somewhat the actual volumetric capacity of a compressor, because, as the entering air is thus warmed, a smaller weight of it is taken into the cylinder per stroke. It has already been shown that the amount of this heating is less in a two-stage than in a single-cylinder compressor.

CHAPTER VII

AIR VALVES

In the earlier years of compressor practice the inlet and discharge valves exhibited wide variations in principle and design. They included spring-controlled poppets, piston valves, various forms of oscillating valves of the Corliss type, under partial mechanical control, together with combinations of poppet and Corliss valves. Of these the simple poppet was the commonest. Typical examples of all of them are given in detail in the previous editions of this book.

Valves of the "thin-plate" type, the first of which was invented about 1894, did not come into general use until nearly twenty years later. They have proved to be so efficient that they have now superseded all others for standard compressors in the United States, although the piston valve is still used by some European makers, as noted later.

Notwithstanding the fact that the poppet and Corliss valves are practically obsolete, a few are to be seen in old-style compressors still in service. For this reason, and because a knowledge of their operation will aid in comprehending the advantages of the "thin-plate" valves, some attention to the older types is given in this chapter. As inlet and discharge valves operate under materially different conditions, they are considered separately.

INLET VALVES

Chief Requisites.—(1) They must have a sufficient area of opening to permit free entrance of the inlet air. (2) They must open with minimum resistance near the beginning of the stroke, remain open until the end of the stroke, and close promptly. (3) They should be of light weight, to operate with minimum noise and shock.

Inlet valves can open only when the pressure within the cylinder falls below atmospheric pressure; that is, when the compressed air occupying the clearance space at the end of a stroke has been re-expanded on the return stroke to a pressure somewhat below that of the external air. To function promptly, the valves must therefore be capable of opening under a slight difference in pressure. From this it follows that the weight of each valve should be as small as is consistent with the strength necessary to withstand the maximum cylinder pressure, when the inlet valves are closed.

The point of stroke at which the inlet valves open depends upon the terminal air pressure at which the compressor is working, and the volume of piston clearance. For a given clearance volume, the higher the terminal pressure, the farther must the piston travel on its return stroke in order to re-expand the clearance air below atmospheric pressure. Not until this takes place can the inlet valves admit a fresh supply of air to the cylinder. They must, therefore, open with minimum resistance. Attention should here again be called to the fact that, although clearance reduces the volumetric capacity of the cylinder, it does not cause loss of work (compare equations 16 and 24, Chapter II). The work expended in compressing the clearance air is recovered; because, simply expressed, the re-expansion of this air on the following stroke aids in overcoming the inertia of the reciprocating parts of the compressor and in compressing the next cylinderful of air.

Inlet Area, or the total net area of the inlet valves should be such as to cause a reasonable velocity and frictional resistance of flow of the entering air. Since the inlet area is properly a function of the piston speed, it may be smaller for slow- than for high-speed compressors. With thin-plate valves, the velocity of flow ranges in the smaller compressors from say 5,000 to 7,500 ft. per min., and in the larger from 4,000 to 5,500 ft. per min. On this basis the inlet area in modern compressors having thin-plate valves varies from about 12 to 20 per cent of the piston area. It was somewhat less in the old poppet-valve compressors.

Diameter and Lift of Inlet Valves.—Poppets, of $2\frac{1}{2}$- to $3\frac{1}{2}$-in. diameter, had a lift of $\frac{1}{4}$ to $\frac{3}{8}$ in. Thin-plate valves, for different sizes and types of compressor, vary in diameter from $1\frac{1}{4}$ to 15 in., their lift being usually from 0.04 to 0.2 in. Small lift obviously minimizes the hurtful effect of inertia, and facilitates prompt opening and closure of the valve. Owing to its peculiar design, the Worthington "feather" valve (Fig. 18) has a relatively high lift; the maximum, in the middle of the individual valve strips, being $\frac{1}{4}$ in. for the smaller sizes of valve, to $\frac{3}{8}$ in. for the larger.

Intake Resistance, the total resistance to the admission of air presented by the inlet valves and valve passages, must be kept as small as practicable. Its effect is to decrease the volumetric capacity of the compressor.

Fig. 1 is a card from the air end of a poppet-valve compressor, showing a total intake resistance of 14.7 − 13.4 = 1.3 lb. per sq.in. The card in Fig. 1*a*, from an Ingersoll-Rand compressor with thin-plate

valves, shows a resistance of less than 1 lb., and may be taken as representing good modern practice. The smaller resistance of thin-plate valves is due to their small lift, weight and inertia, and, because of their relatively large size, fewer valves are required.

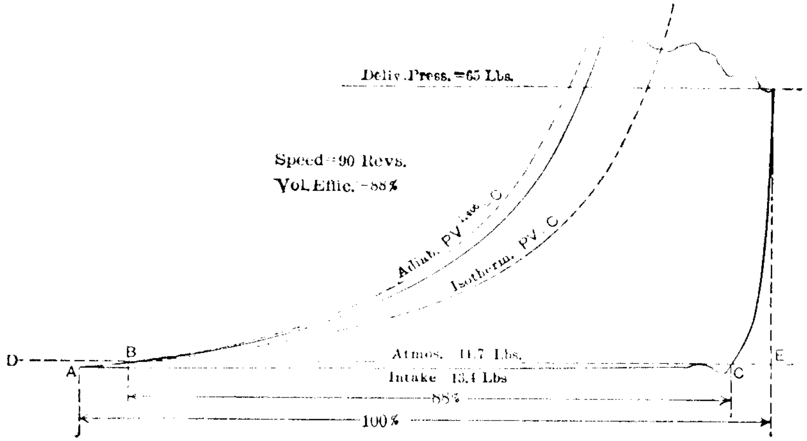

Fig. 1. Effect of Intake Resistance in a Poppet-valve Compressor.

Referring to Fig. 1, *AC* is the admission line, *DE* the atmospheric line, and the area of the card between them measures the loss of work due to intake resistance. Point *B*, where the compression line crosses the atmospheric line, is the point of the stroke that must be reached by the piston before fresh atmospheric air can enter the cylinder. The volume passed through by the piston in traveling from *A* to *B* represents the loss of volumetric capacity caused by intake resistance only. The total loss of capacity, including that due to piston clearance, is measured by the length of $AB + CE$; and the net volumetric efficiency of the compressor is equal to *BC* divided by the total length, *AE*, of the

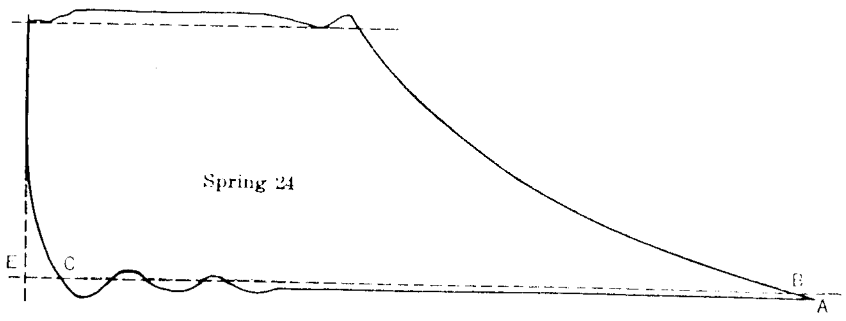

Fig. 1*a*.—Indicator Card from Air End of a Compressor Having Thin-plate Valves.

indicator diagram. **In Fig. 1** the volumetric **efficiency is 88 per cent**; in Fig. 1*a*, 95 per cent.

The rate of flow of the intake air into the cylinder varies with the piston speed of a given compressor. When the speed is greatest, in the middle of the stroke, the rate of inflow is at the maximum. While the piston is moving slowly, near the beginning and end of each stroke, as the crank approaches and turns its centers, the relatively small negative pressure may be insufficient to keep the valves open constantly. This irregularity is greater with the spring-controlled poppets than with thin-plate valves.

Poppet Valves, though almost obsolete for single-stage compressors, are still used in a modified form for the high-pressure cylinders of multi-stage compressors.

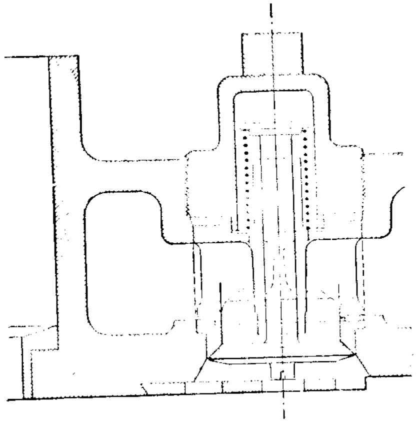

Fig. 2. Norwalk Poppet-valve (old style).

A common form of poppet is the mushroom valve (Fig. 2), for which there are two special requirements: (1) the area of each valve must be small, or the valve will be too heavy, injuring its seating surface, and by its inertia causing excessive resistance to the spring; (2) the lift must also be small, to insure prompt opening and closure. Hence, the total inlet area required must be furnished by a group of valves in each cylinder head.

The valve stem is encircled by a spiral brass spring, easily compressible and accurately calibrated and adjusted to minimize resistance to opening. Fig. 3 shows the losses of volumetric capacity caused by different valve-spring resistances. At altitudes above sea-level the loss increases, because the piston must travel farther on its return stroke (between *E* and *C*, Fig. 1) to re-expand the clearance air below atmospheric pressure. As the springs tend to close the valves against the atmospheric pressure, irregularities occur in the entrance of air, producing "dancing" or "chattering" of the valves on their seats. If the springs are too slack, chattering increases; if too tight, the valves open later in the stroke.

As already stated, special modified poppet valves are still used in most multi-stage compressors, for dealing with the heavy pressures in the cylinders after the second stage, even when some other type is preferred for the low-pressure cylinder. Except for such service, poppet valves are obsolete.

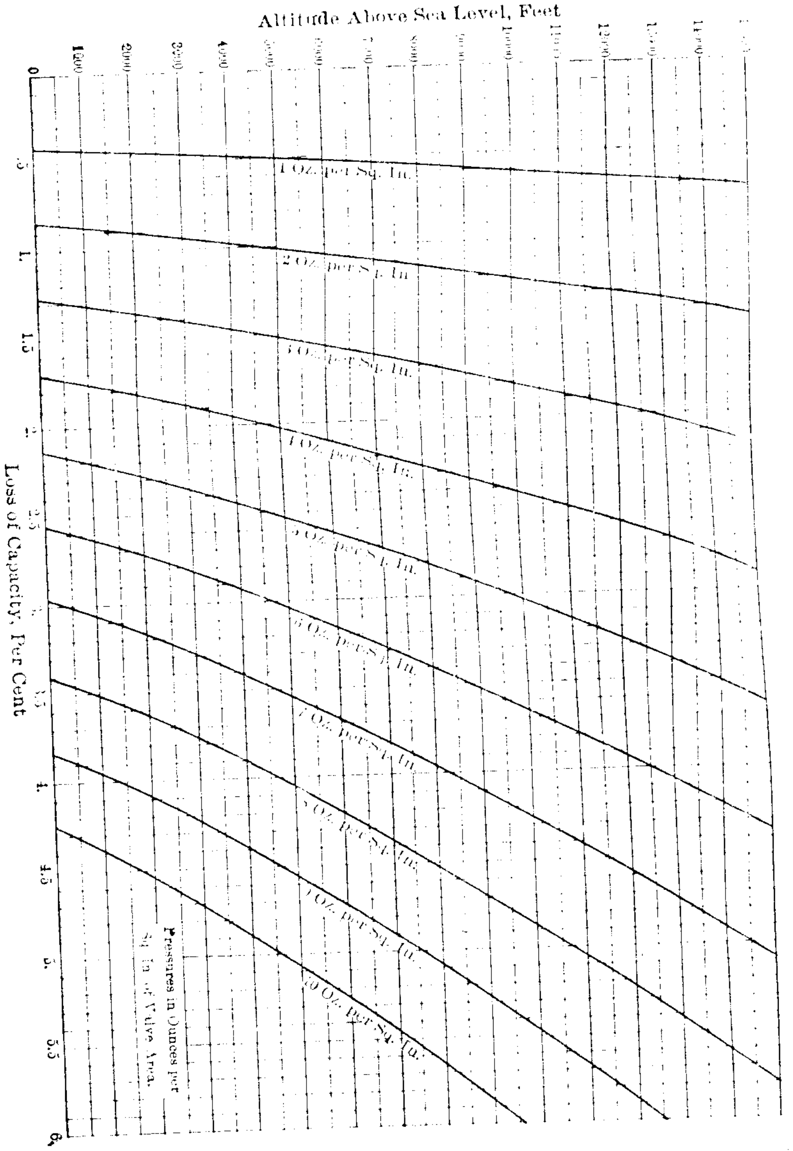

Fig. 3.—Effect of Spring Resistance of Poppet-valves on Volumetric Capacity of Compressors (*Eng. News*).

Positively-operated Inlet Valves.—These are not actuated by difference between the air pressures inside and outside of the cylinder, but by rods or links connected to an eccentric on the compressor crank-shaft. They therefore open and close at fixed points of the piston's stroke, uninfluenced by springs. See also under "Mechanical Control of Air Valves," near end of this chapter.

The piston valve of W. H. Bailey & Co., Manchester, England, successfully employed in one of their lines of compressor, is an example of this type. It was developed from the Köster valve, used by some compressor builders on the Continent of Europe. Only one inlet valve, of large area, is required at each end of the cylinder, both being mounted on a common longitudinal rod. Fig. 4 shows this valve applied to a single-stage cylinder. *A*, *A* are the inlet valves and *B*, *B*, the discharge valves; *C*, *C*, cylinder ports; *D*, air inlet; *E*, *E*, discharge chambers. Fig. 5 exhibits the arrangement of valves in the two-stage differential

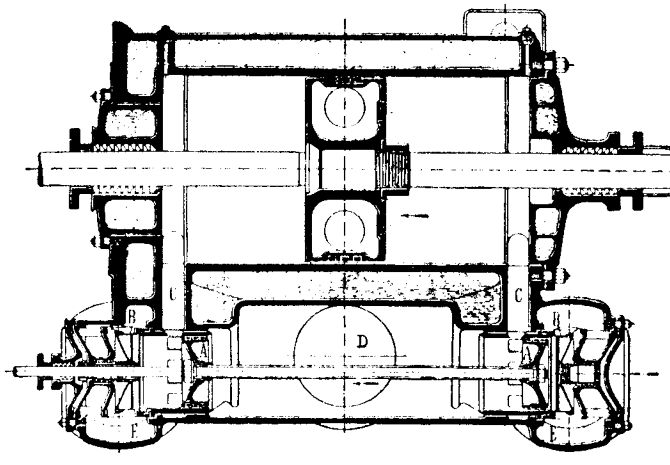

FIG. 4.—Bailey Piston Valves for Single-stage Compressor.

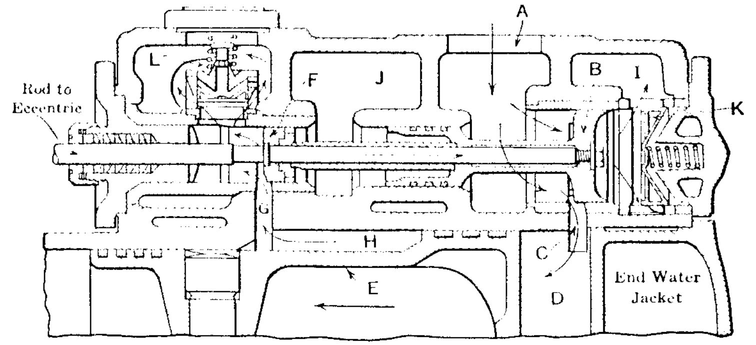

FIG. 5. Bailey Piston Valves for Differential, Two-stage Compressor.

compressor of the same maker. *A* is the air inlet; *B*, low-pressure inlet valve; *C*, port to low-pressure end of cylinder (*D*); *E*, differential piston; *F*, high-pressure inlet valve; *G*, port to high-pressure end of cylinder (*H*); *I*, connection to intercooler; *J*, inlet from intercooler to high-pressure end; *K*, *L*, low- and high-pressure delivery valves. The arrows indicate the course of the air during the stroke from right to left. This valve is so shaped that it has a very small clearance space as it advances close to the delivery valve; it is simple in construction, operates quietly, even at high speeds, and permits use of cylinder water jackets of large area.

The piston valve illustrates the principle of complete mechanical control, as distinguished from valves under partial mechanical control, like the Corliss and Riedler valves noted below.

Obsolete Inlet Valves.—Several of these deserve brief mention here. The Corliss cylindrical, oscillating valve, an adaptation of the well-known steam Corliss valve, was formerly widely used, as in the Norwalk, Allis-Chalmers, Nordberg and other compressors. The Riedler valve differed in principle from other mechanically-controlled valves in an important particular. At the beginning of the stroke, the inlet valve was free to open under difference of air pressure. Near the end of the stroke, as the piston's velocity decreased, the valve was caused to approach very close to its seat (about $\frac{1}{16}$ in.), being thus prepared for prompt closure by the mechanical control as the stroke reversed. Still other forms of inlet valve were employed by different makers, and are to be seen in some old compressors in service. Among them is the Ingersoll-Rand "hurricane" inlet, consisting of a single large ring-valve, working in guides in each face of the piston and actuated by its own inertia as the piston reciprocates.

For full details of these mechanically interesting valves see the previous editions of this book.

THIN-PLATE VALVES

These, also known by the trade names of "feather," "wafer" and "plate" valves, have superseded all other types in standard compressor practice, for both inlet and discharge.

As stated near the beginning of this chapter, for attaining maximum volumetric capacity of a compressor, the inlet valves must open promptly, with minimum resistance and at the earliest possible point of the stroke. These requirements are imperfectly met by poppet valves, which originally and for many years were almost exclusively employed for compressors. When used as inlet valves, the disadvantageous effects of inertia are clearly exhibited; also, their springs are subject to frequent changes in resistance, and require careful calibration and adjustment and constant attention. The "chattering" and noise of poppets are evidences of irregularity in the admission of the intake air. It is probable that these imperfections first suggested the advantages attainable by using thin-plate valves of very light weight.

The chief objects sought in introducing the thin-plate valves may be summarized as follows: (1) to secure greater simplicity; (2) by decreasing the weight of each individual valve, to minimize the effects of inertia; (3) to increase the effective area of each valve, thereby reducing their number; (4) to permit the use of water-jackets of larger area, especially on the cylinder heads, where cooling is most efficacious.

Early Types.—Among the first thin-plate valves were the Gutermuth, Rogler, and Leyner. The Gutermuth valve (now of historical interest only) consisted of a rectangular steel plate and a grid seat. One side of the plate was coiled in a spiral, like a clock spring, around a stationary spindle, the inner edge of the spiral being inserted in a longitudinal groove in the spindle. By placing several valves side by side, any desired area of opening was obtainable. The valves opened against the slight resistance of their springs. The Rogler valves, first applied about 1891 on compressors at the Simplon tunnel, consisted of a flexible steel annulus with several concentric port openings. In its present modified form, it is known as the Rogler-Hoerbiger, one of the best of the thin-plate valves. The Leyner valve, invented about 1905, and illustrated in former editions of this book, is now obsolete, but its fundamental principle has been influential in the designs of some of the later thin-plate valves.

Although all the thin-plate valves in current use embody similar general features, they differ in their details. The seating portion of some is a single steel plate of various shapes; others consist of one ring (annulus), or two or three separate, concentric rings. All have grid seats. A further classification, comprising nearly all designs shown in the following illustrations, will aid in comprehending their differences: (1) valves that have one or two large concentric spiral springs (Allis-Chalmers Co., American Air Compressor Works, Holman Bros.); (2) those having groups of small helical springs (Belliss & Morcom, Chicago Pneumatic Tool Co., Ingersoll-Rand Co., Norwalk Co., Pennsylvania Pump and Compressor Works); (3) those in which the spring element is integral with the valve itself (Walker Bros.). Besides the above, there are the two unique designs of the Sullivan Machinery Co. and the Worthington Pump and Machinery Corp.

Operation of Thin-plate Valves.—Because of its light weight, the inertia of the seating portion of the valve is small. At the beginning of the stroke, when the air in front of the piston is approximately at atmospheric pressure, springs of very slight resistance will close the valves promptly and with minimum shock. Above the valve is a cushion- or stop-plate, to limit its lift. For valves of large area, the lift is generally about $\frac{1}{8}$ in. (occasionally less); for those of small area, from $\frac{1}{4}$ to $\frac{3}{8}$ in. In some designs, a guard or guide plate keeps the valve in position for accurate closure over the grid-seat ports; this plate has recesses to hold the springs in place. All the valve parts are assembled on a central spindle or bolt, which, in turn, is often screwed into the cover plate on the cylinder casting. By taking off the cover plate, the valve is readily removed for inspection or cleaning.

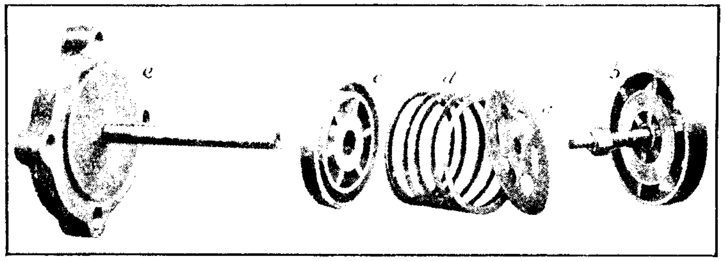

FIG. 6.—Allis Chalmers Valve.

Following are descriptions in alphabetical order of type forms of thin-plate valves, for further illustration of their construction and operation.

Allis-Chalmers M'f'g Co. (Fig. 6). This is one of the simplest designs of thin-plate valve. The seating portion is a single steel ring plate *a*, working between the grid seat *b* and the guard plate *c*, under control, of the spring *d*; *e*, the cover plate for the valve chamber, has a hollow stud fitting on the end of the seat bolt.

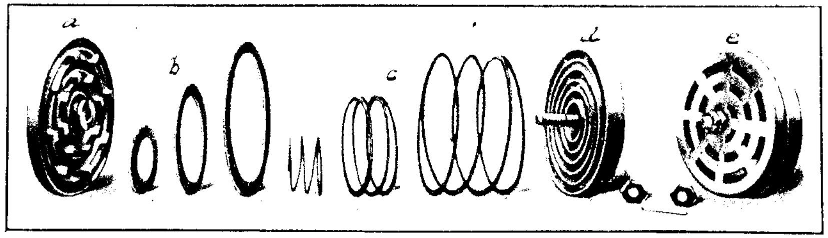

FIG. 7.—Valve of the Etablissements François.

Somewhat similar to the above are the "Fairhurst" valve (**American Air Compressor Works**), which is attached to the seat by two small bolts, and the valve of **Holman Bros.**, except that the latter has two concentric spiral springs, instead of one, and is held in position by two studs.

Etablissements François (Fig. 7).—This valve, which resembles the Norwalk (see below), except for the design of the springs, comprises: seat *a*, three very light valve rings *b*, three spiral springs *c*, and guard or stop-plate *d*; *e* showing the assembled valve. The valve rings are held in position, as they open and close, by two concentric series of studs, cast on the seat. Fig. 8 shows two inlet and two discharge valves in place on the cylinder of Type B compressor of this maker.

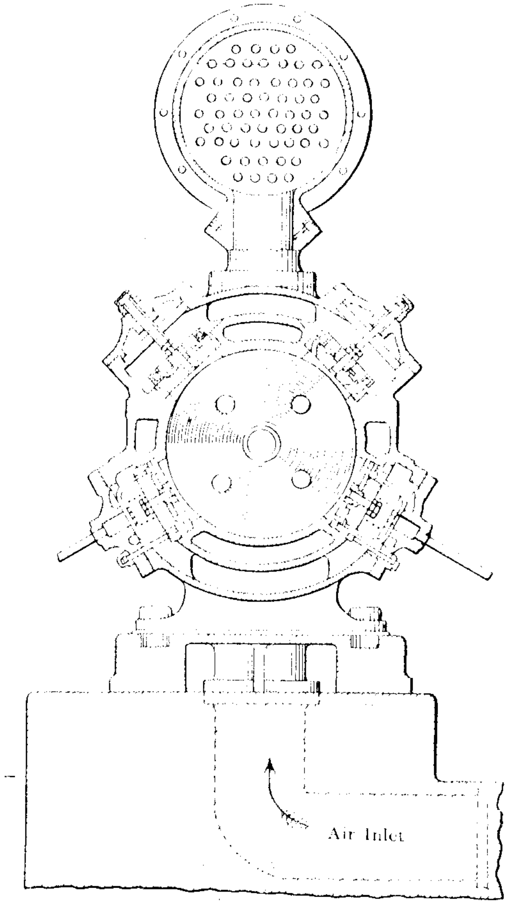

Fig. 8.—Valves in Position. Etablissements François.

Ingersoll-Rand Co. for a number of years used the Ingersoll-Rogler thin-plate valve. In this the closing spring formed part of the cushion plate (for details, see the third and fourth editions of this book). All their compressors are now fitted with a valve which combines the essential features of the Rogler-Hoerbiger (see below) and the old Leyner valve.* The design of valve for the XPV compressors of this company is shown by Fig. 9. It consists of a grid seat, and a perforated steel disk retained in position by a guide-plate, above which is a stop-plate, limiting the lift of the valve and having recesses for the springs. A washer between the valve and seat determines the height of lift (about ¼ in.). The dowel-pin, close to the central bolt, prevents the valve from turning, and causes it to seat always in the same position. Fig. 10 shows the top and bottom views of an assembled valve, of slightly different design, as used in the Type PRE compressors.

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[*] The Ingersoll-Rand Co., some years ago, acquired the patents and business of J. Geo. Leyner, of Denver, a pioneer in the United States in introducing the thin-plate valve.

Norwalk Co.—Fig. 11 shows the form of valve for the first and second cylinders of a stage compressor. It comprises: grid seat *a*; two or three valve-rings *b*; four light helical springs *c*, for each valve ring; guard-plate *d*, with studs for holding the springs in position; retaining-gland *e*, cover-plate *f*. Fig. 12 is the cross-section of an intake cylinder, with the valves in place. Fig. 13 shows the design of the "ring-plate valve" for the third and fourth cylinders of multistage compressors. The seating portion is a single heavy disk, closed by a strong helical spring. The seat contains six circular ports.

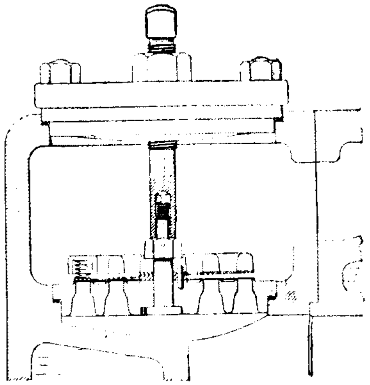

Fig. 9. Ingersoll-Rand Valve in Position. Form Used in Type XPV Compressor.

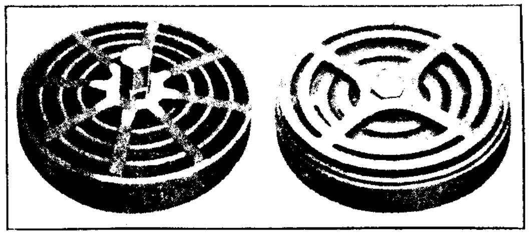

Fig. 10.—Ingersoll-Rand Valve for the Larger Sizes of Type PRE Compressors. (Top and bottom views.)

(Note.—In the older compressor designs of this company, made only to order, ordinary poppet valves are used.)

The "Simplate" valve, of the **Chicago Pneumatic Tool Co.,**

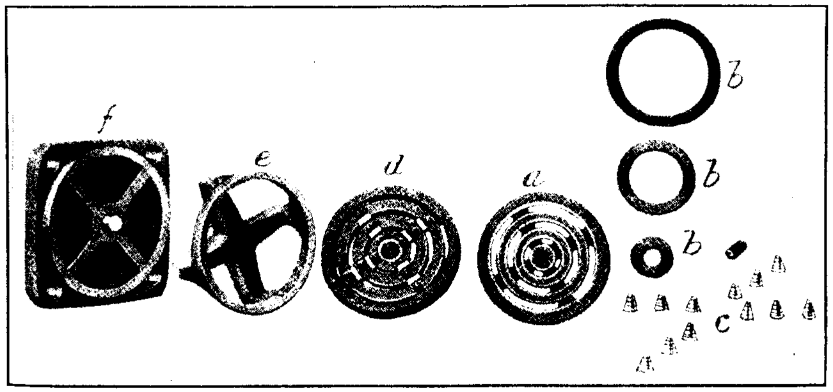

Fig. 11.—Norwalk Valve, Low-pressure Type.

resembles the Norwalk in general design. It has two concentric seating rings, each closed by three helical springs.

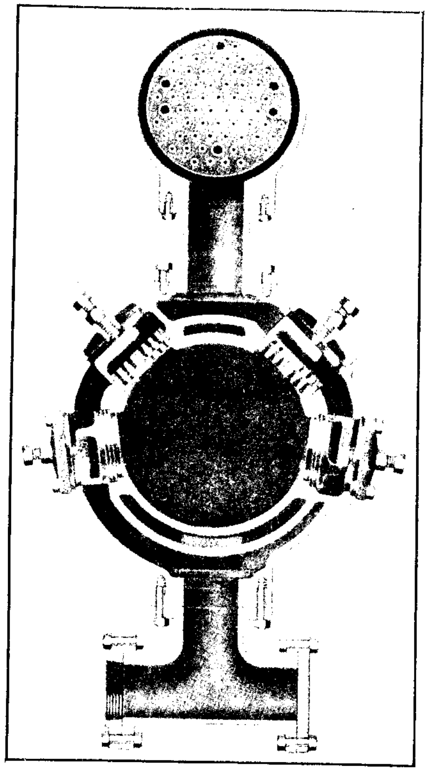
FIG. 12.—Norwalk Intake Cylinder, with Valves in Place.

The valve of the **Pennsylvania Pump and Compressor Co.** (Fig. 14) has three seating rings, and is also similar in principle to the Norwalk.

Rogler-Hoerbiger Valve (Fig. 15), a modification of the Rogler, already referred to, is used in all the compressors made by **Belliss & Morcom.** By comparing this valve with many of the others, it will be seen that its essential features have been widely adopted in the design of thin-plate valves. It comprises the grid seat (1) having three concentric series of ports, guard-plate (2) with four recesses for the springs (3), valve (4), lift washer (5), and cushion or stop-plate (6). The assembled valve is shown by (7). As this valve is of large diameter, its lift is only about $\frac{1}{16}$ in., approximating the thickness of the valve-plate. The seating portion of the valve is connected in a flexible manner to its fixed center (through which passes the bolt) by means of the inner curved arms (8), which are cut through diagonally in two places. This causes the valve to seat always in the same position, while still insuring freedom of movement in opening and closing, and a uniform lift over the whole area of the valve.

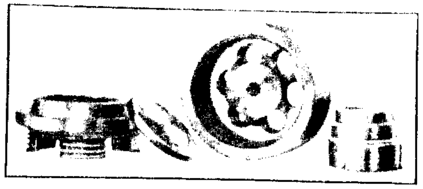
FIG. 13.—Norwalk "Ring-plate" Valve, High-pressure Type.

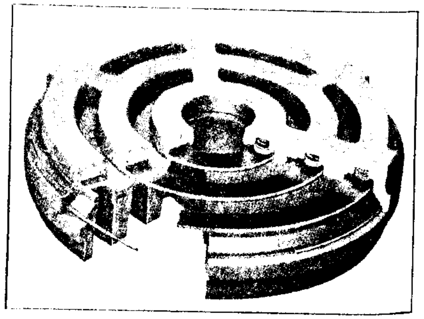
FIG. 14. Plate Valve and Seat. Pennsylvania Pump and Compressor Co.

The seating portion of the discharge valves of the quadruplex compressor of **Reavell & Co.** (Fig. 12, Chapter III) somewhat resembles that of the Rogler-Hoerbiger valve; but there are two series of concentric ports, instead of three, and the valve is held on its seat by several curved, radial spring arms, the inner ends of which are fastened by the central bolt of the valve. As this compressor has single-acting cylinders, no suction valves are required. The inlet air is admitted through ports in the connecting-rod gudgeons, which communicate with slots in the pistons.

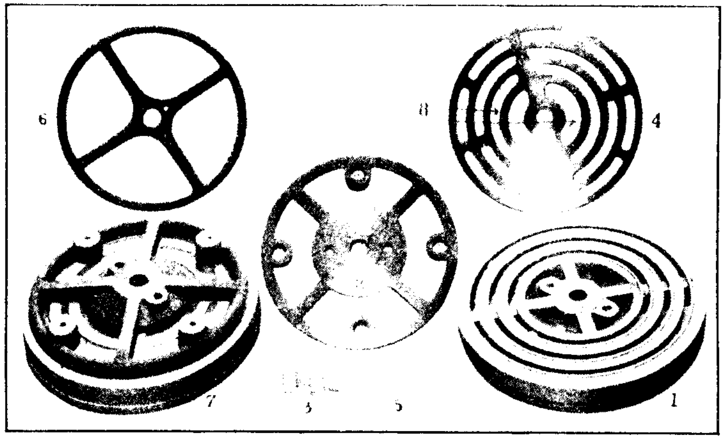

FIG. 15.—Rogler-Hoerbiger Valve. (As used by Belliss & Morcom.)

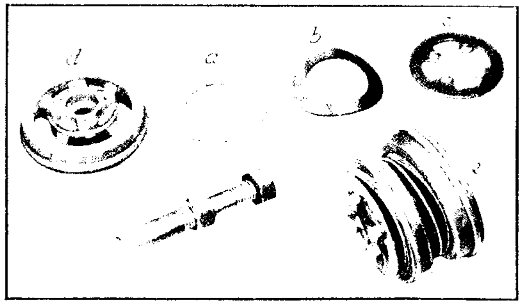

FIG. 16.—"Wafer" Valve, Sullivan Machinery Co.

Sullivan Machinery Co. formerly used a "multiple-finger" thin-plate valve, details of which are given in the former editions of this book. This was superseded about 1918 by the "wafer" valve (Fig. 16). It consists of a single ring *a*, of small diameter. The closing spring *b* is made of a ring of the same size as the valve, but has a double curvature.

as shown. Above the spring is the guard *c*; *d* is the assembled valve. The concave side of the spring is set against the valve, the convex side against the guard.

In the "angle-compound" compressors of this company the inlet and discharge valves are placed in tandem in each port (see *c*, Fig. 16) This construction simplifies the cylinder casting and each pair of valves can be taken out together for inspection or cleaning, by removing a screw plug.

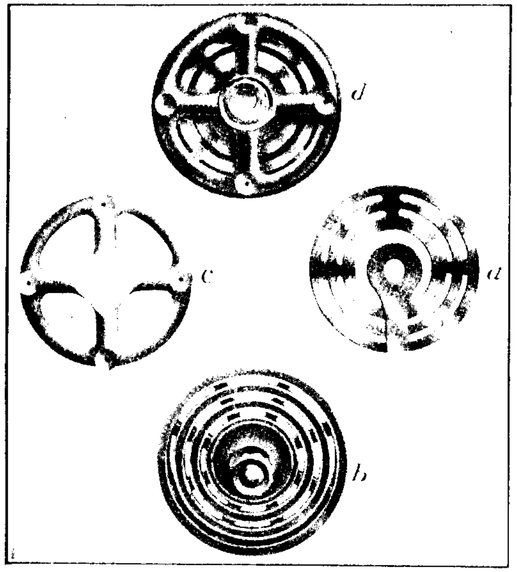

Fig. 17.—Walker Bros. Thin-plate Valve.

Walker Bros. valve (Fig. 17) consists of the perforated seating plate *a*, grid seat *b* and valve guard *c*. The bottom and top of the assembled valve are shown respectively at *b* and *d*. Closure is effected by the radial arm, the inner end of which is fixed to the central bolt and the outer end riveted to the periphery of the valve plate. Thus the spring element is integral with the valve itself, no independent springs being used.

Worthington Pump and Machinery Corp.—The valve used by this company, formerly known as the Laidlaw-Dunn-Gordon "feather valve,"* has been in satisfactory service for many years. In Fig. 18 the seat is on the right, the valve cover, with its curved guard strips, on the left. The seating portion consists of five thin rectangular strips of steel. Their ends, though not rigidly attached, do not lift from the seat, but are retained in position by the rim of the guard plate. The middle portion of each strip opens against the convex surface of the guard above it, and on closing seats itself quietly, by contact not impact, over its slot in the seat. There are no springs, and the effect of inertia is almost *nil*, although the lift is comparatively high. This valve is identical in construction and size for both inlet and discharge. It is shown also, with its retaining yoke and set-screw, in Fig. 37, Chapter III. Sizes of valve range from 4 in. diameter for a 7-in. cylinder to 11¼ in. for a 40-in. cylinder. The number of "feather strips" depends upon the diameter of the valve.

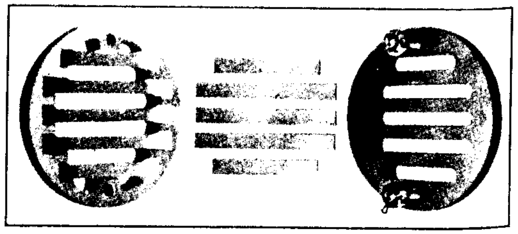

Fig. 18.—Worthington "Feather" Valve.

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* Some years ago, the Worthington Corporation acquired the business of the Laidlaw-Dunn-Gordon Co.

Arrangements for Admitting Inlet Air.—The colder the intake air the smaller the volume occupied by a given weight of air taken into the cylinder, and the greater the volumetric output. Frank Richards states:* "The volume of air at common temperatures varies directly as the absolute temperature. With the air supply at 60° its absolute temperature is 521°, and its volume will increase or decrease $\frac{1}{521}$ for each degree of rise or fall of temperature. Therefore, if in securing the supply of air we can get a difference in our favor of 5° . . ., we accomplish a saving of about 1 per cent. If a difference of temperature of 10° can be secured 2 per cent is saved," practically without cost (see Table 1). The practice of taking air from the engine-room, common at mines, is bad; such air is usually quite warm, and is apt to be charged with dust, which causes unnecessary wear of valves and piston. Fresh air should be taken from some point outside of the compressor house. A wooden or concrete conduit is better than one of iron, because of its smaller conduc-

Table 1.—Effect of Intake Temperature on Compressor Capacity

(Normal capacity assumed to correspond to 60° F.)

Initial tempera-ture, deg. F.	Absolute temperature, deg. F.	Relative capacity	Initial tempera-ture, deg. F.	Absolute temperature, deg. F.	Relative capacity
−20	440	1.18	80	540	0.96
−10	450	1.16	90	550	0.94
0	460	1.13	100	560	0.93
10	470	1.10	110	570	0.91
20	480	1.08	120	580	0.90
30	490	1.06	130	590	0.88
40	500	1.04	140	600	0.86
50	510	1.02	150	610	0.85
60	520	1.00	160	620	0.84
70	530	0.98			

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* "Compressed Air Practice," p. 67.

tivity. Its cross-section should be at least one-half the area of the cylinder, to minimize frictional resistance. Table II gives recommended sizes of conduit for different volumes of free air compressed per minute.

Table II. Sizes of Intake Conduits

(Peter Brotherhood, Ltd., Peterborough, England)

Free air per min., cu.ft.	Diam. of duct, in.	Side of square duct, in.	Free air per min., cu.ft.	Diam. of duct, in.	Side of square duct, in.
40	2	1¾	700	12	10½
60	3	2⅝	900	14	12½
80	4	3½	1,200	16	14
100	5	4⅜	1,500	18	16
140	6	5¼	2,000	20	17½
200	7	6¼	2,500	22	19½
300	8	7	3,000	24	21½
500	10	9			

For convenience in connecting the intake, the inlet valves may be enclosed in external air chests on the cylinder heads. By these means a greater saving can be effected in winter than in summer, but even in warm weather something is gained, especially if the intake conduit opens on the north side of the building, out of reach of the sun's direct rays, and is carried to some height above the ground.

Care should be taken to prevent the entrance of dust or rubbish. If the intake is open, particles floating in the air may be drawn in and obstruct the valves or injure their seats and the working surfaces of piston and cylinder. Dust tends to cling to the oil film on the cylinder walls; too much oil collects more dust, too little means insufficient lubrication.

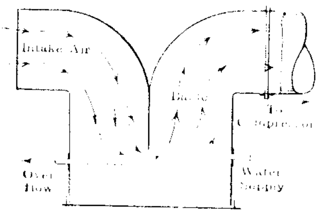


Fig. 19.—Baffle for Removing Dust from Intake Air.

Fig. 19 shows a baffle device for intercepting dust; the surface of the water must be kept clean by a small continuous flow, as indicated. Another air cleaner is shown by Fig. 20. It is a light wooden box, with removable light frame panels, which are covered with ½-in. mesh wire screen, and to which cheesecloth or loosely woven flannel is attached. The cloth is cleaned or replaced at intervals. The screen area should be at least 1 sq.ft. for each 25 cu.ft. per min. of free air capacity of the compressor. In hot climates, the cloth may be kept wet, to cool as well as cleanse the air. A cobblestone cleaner has also been recommended. It consists of a box, say 18 in. wide by 36 in. high, with several shelves of strong wire mesh, on which are placed layers of washed stones, coated with heavy oil. The dust adheres to the stones, which are periodically removed and cleaned.

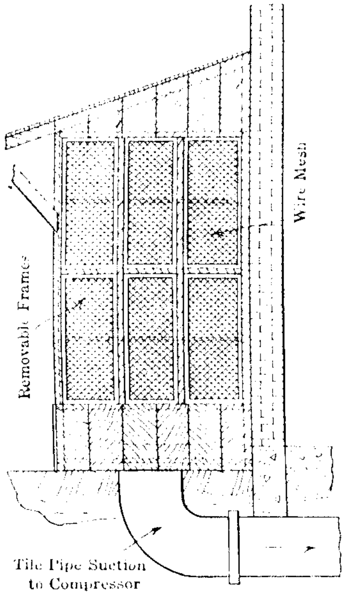


Fig. 20.—Intake Box, with Cheesecloth Screens. (Trade Standards, Compressed Air Soc.)

Discharge Valves

General Considerations.—While inlet valves must operate under small differences of pressure, discharge valves are subjected to heavy pressure on both sides. Furthermore, owing to irregularities in air consumption, the receiver pressure fluctuates, so that the point of stroke at which the discharge valves open depends not only on the ratio of compression at which the compressor is working, but also on the relation between the variable pressures in cylinder and receiver. Hence, for discharge valves, the sensitiveness of operation essential for inlet valves is unnecessary.

Chief Requisites.—Discharge valves must be strong enough to withstand the maximum cylinder pressure, have strong springs to close them promptly at the end of the stroke, fit accurately on their seats to prevent leakage, and have ample guiding surface to insure accurate alinement in their movements. Delay in closing, or leakage between valve and seat, are far more serious than with inlet valves. Leakage of even a small quantity of compressed air back into the cylinder is equivalent to the volumetric loss caused by an excessively large clearance space.

Notwithstanding the differences between the operating conditions of inlet and discharge valves, stated above, the importance of these conditions has been greatly lessened by the introduction of the thin-plate valves. The types of poppet discharge valves, together with several special designs, are set forth at length in previous editions of this book. In current practice, inlet and discharge thin-plate valves are of similar design; in many cases they are identical. It is therefore unnecessary to repeat their details here.

Discharge Area.—The volume of air in the cylinder having been reduced to a small fraction of its initial volume, it might appear that the total discharge area could be made smaller than the inlet area, without causing excessive frictional resistance. But, discharge must take place in a relatively short period of time. Although the inlet valves are open throughout a large part of the stroke, delivery takes place while the piston is making the last third or half of its stroke. Therefore, the discharge area is generally about equal to the inlet area; that is, for compressors with thin-plate valves, it is from 12 to 20 per cent of the piston area (see previous discussion under "Inlet Valves").

The above considerations apply also to the area of the passages from the valves to the pipe leading to the receiver. When the valves first open, the piston is moving at nearly maximum velocity. If the velocity of discharge is high enough to cause large frictional resistance during the delivery period, the cylinder pressure will rise momentarily above the normal, and then drop back after the air has passed out. As equilibrium with the receiver pressure is not attained until the piston is close to the end of its stroke, loss of work ensues.

Another source of loss may be explained as follows: For all automatic valves, the area of the seating surface should be no greater than is necessary for durability. This is especially important for discharge valves, because of the high pressures under which they act. Thus, in compressing to 90 lb., assume the total area of the valve on the receiver side to be 10 sq. in., and the net area of the opening in the seat to be 8 sq. in. Then, the total pressure tending to keep the valve closed will be 10 × 90 = 900 lb., and, before the valve can open, the cylinder pressure must rise theoretically to 900 : 8 = 112.5 lb. per sq.in. However, as the seating surface is never perfect, the valve actually opens at a lower pressure—say 105 lb. As the work of compressing to this momentary increased pressure of 105 − 90 = 15 lb. is useless work, the seating surface should be as small as is consistent with durability.

The total amount of the above losses is represented by the irregular area of the air card (see Fig. 1) lying above a horizontal line drawn through the point corresponding to the pressure at the end of delivery.

MECHANICAL CONTROL OF AIR VALVES

Before the general adoption of thin-plate valves, mechanical control of air valves in a number of different forms was quite common. It was employed, for example, in all compressors having valves of the Corliss type; besides which were several designs involving rather complicated special valve motions, like the Riedler, Nordberg, Laidlaw-Dunn-

Gordon, and Allis-Chalmers. For details of these practically obsolete designs, see Chapter IX of the fourth edition of this book. A brief statement only will be made here.

Principles.—By mechanical control, the action of the valves is in some manner connected with the rotary or reciprocating parts of the compressor. The operating effect of the valve springs is thus so modified as to cause the valves to open and close more promptly. Hence, fewer valves are required, because each could be made larger and have a higher lift; in some designs there were only one inlet and one discharge valve at each end of the cylinder (for example, the Riedler).

There are two general classes of mechanical control. In one, the mechanism is timed to act at fixed points of the piston stroke, the valves being actuated positively. This principle is exemplified by the Bailey valves (Figs. 4 and 5, with the accompanying text). In the other type, the control is partial only. It causes the valves at the proper time to approach close to their seats, but leaves them free to act finally under difference of air pressure; thus allowing for variations in working conditions, inaccuracies of adjustment, or slight derangements due to wear of parts.

In general, mechanical control has been found best applicable to inlet valves of single-stage compressors, or the low-pressure cylinder of multistage compressors, because their opening depends primarily upon small differences in air pressure. They must open at the instant when the clearance air has re-expanded slightly below atmospheric pressure. The inlet valves of the high-pressure cylinders, being subjected to heavy pressure on both sides, are best allowed to open and close automatically.

Mechanical Control of Discharge Valves is unsatisfactory, for the reasons set forth above and under "General Considerations" affecting the action of discharge valves. In attempting to open them mechanically, at a fixed point of the stroke, two cases may occur: (1) if the receiver pressure drops below normal, the controlling mechanism might be injured, due to excess of cylinder pressure before the fixed point of opening is reached; (2) if the receiver pressure rises above normal, the valves would be held forcibly on their seats, by the excess of receiver pressure, after being released by the controlling gear. In either case, there may be derangement or breakage of some part of the mechanism. To allow discharge valves to adjust themselves automatically to varying working conditions, they must have some degree of freedom as to their time of opening. A number of devices have been introduced for dealing with these conditions; for example, relief valves operating in conjunction with mechanically-controlled discharge valves.

CHAPTER VIII

SPEED AND PRESSURE REGULATORS AND UNLOADERS

THE conditions under which compressors usually operate cause wide variations in the demand for air. This is especially true in mining and general rock excavation. Variations are apt to be greatest when a single compressor furnishes air for only a few rock-drills; conversely, the larger the number of drills operated, the more uniform is the consumption (see Chapter XX, under "Consumption of Air"). The compressor speed must therefore be adjusted to the rate at which the air is used. When consumption falls below normal, speed is reduced, to prevent the direct loss that would be caused by blowing off air at the receiver. If consumption ceases entirely for a time, the compressor may either be stopped, or be automatically "unloaded" so as to require only enough power to turn its centers.

To meet these conditions, without necessity for constant attendance of an engineer, numerous regulating mechanisms have been devised, whereby the compressor can operate automatically under large variations of load. The examples cited below illustrate the principles and details of the devices employed by the leading compressor builders.

Regulators may be classified as: (1) speed and pressure governors, for steam-driven compressors; (2) air-cylinder unloaders, for belt- and motor-driven compressors. The second class includes intake unloaders and selective or step controls.

GOVERNORS FOR STEAM-DRIVEN COMPRESSORS

With fairly constant air consumption, a steam-driven compressor requires no more speed regulation than is furnished by a governor similar to those of ordinary steam engines, for taking care of fluctuations in boiler pressure, or derangement or breakage of some part of the compressor. But, when the air consumption decreases or ceases entirely, the direct loss due to blowing off air becomes important; because, as demonstrated near the beginning of Chapter XVIII, the power cost of a cubic foot of air is greater than that of a cubic foot of steam. Therefore, the quantity of steam admitted to the cylinder must be coordinated to the variable pressure in the air receiver.

Steam-driven compressors generally have some form of flyball governor, the action of which is modified by the varying receiver pressure. The compressor speed and the air pressure are thus interrelated, without at any time "unloading" the air cylinder. In most designs compressed air, transmitted from the receiver through a pipe, acts on the piston of a small cylinder forming part of the regulating mechanism. The movements of this piston control the radius of rotation of the flyballs and therefore their rotative speed; which in turn adjusts the amount of opening of the steam throttle, or the action of the valves of the steam cylinder. The following examples are in alphabetical order.

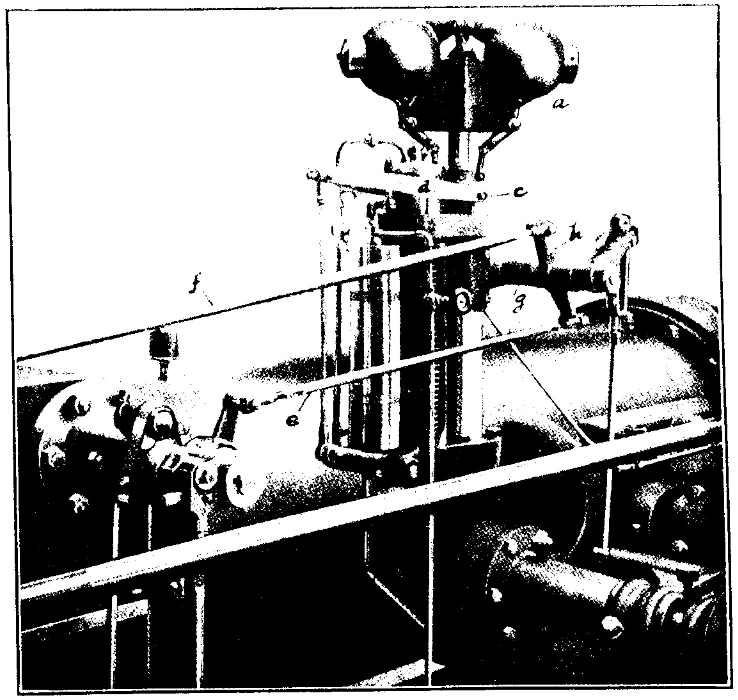


FIG. 1.—Allis-Chalmers Speed and Pressure Regulator.

Allis-Chalmers Man'f'g Co.—Fig. 1 shows the regulator used for a compressor driven by a Corliss engine. The flyball governor *a* is driven through inclosed gearing at *b* by a belt from the compressor crankshaft. Its operation is modified by the "floating lever" *c*, the fulcrum of which, at *d*, is raised or lowered in accordance with the variations in receiver pressure. These movements of lever *c* cause the vertical spindle of the flyballs to rise or fall and alter their rotative radius. A toothed rack on the spindle, engaging with a pinion on shaft *g*, moves the two-armed lever *h* through a small arc, thus causing rods *e* and *f* to alter the points of cutoff of the steam valves. This varies the compressor speed, for maintaining a nearly constant delivery pressure. The makers state that cards from a recording gage show that less than 2 per cent variation in air pressure will vary the compressor speed from full to practically no speed.

Chicago Pneumatic Tool Co., for their steam-driven compressors, use the Jarecki speed and pressure governor (Fig. 2). The split-ball governor (11), belt-driven from the crank-shaft to the pulley (20), accompanied by the tightener (43), controls the steam throttle (3). Connected with the governor spindle and throttle-valve stem (28), is a lever (25), the position of which is influenced by the combined action of the springs (31, 32, and 26). By screwing up or down the handwheel and speeder screw (5), this system of springs (and with them the governor) is set to run the compressor at any desired speed. The other element of the governor is the air-pressure device, which, by the position of the plunger in the small air cylinder (18), brings the springs into action

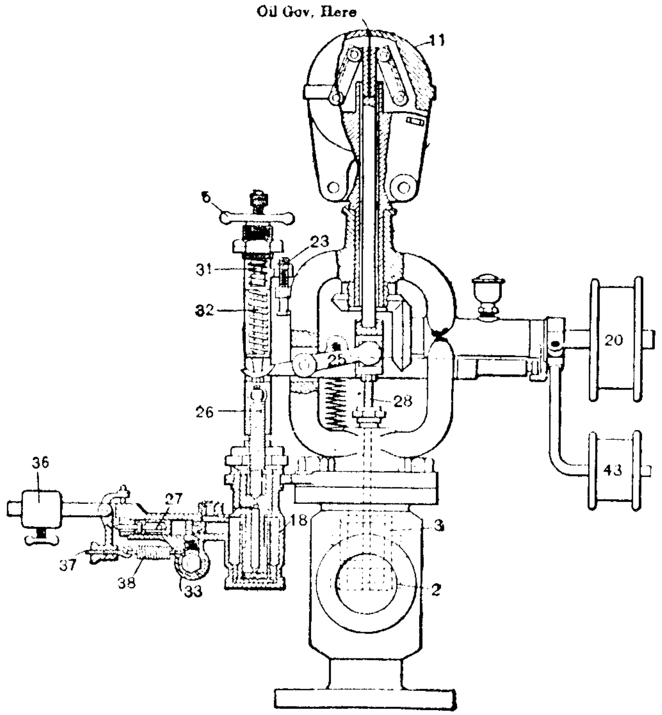


FIG. 2.—Speed and Pressure Governor (Chicago Pneumatic Tool Co.).

in the order of their strength, thus producing movement of the lever (25). Cylinder (18) is connected to the receiver by the union valve (33), which first admits air to the little cylinder (27), the piston of which operates a needle valve. This valve is held closed against any desired minimum air pressure by the adjustable weight (36) and the regulating screw and spring (37 and 38). When the receiver pressure exceeds the normal, the needle valve admits compressed air to cylinder (18). As the pressure increases, the plunger in (18) rises against the counterspring (26) and through the lever (25) tends to close the steam throttle (3), thus slowing the compressor. Total stoppage is prevented by screwing down the nut of the stop-screw (23), so as to limit the upward movement of the plunger (18), which is so proportioned that a variation of only 2 or 3 lb. receiver pressure is multiplied to say 40 lb. in its action on the governor. A sensitive control is thus produced within narrow limits of working pressure. To prevent violent movements of the pressure element, in case of sudden changes in receiver pressure, the plunger in (18) has an oil dash-pot.

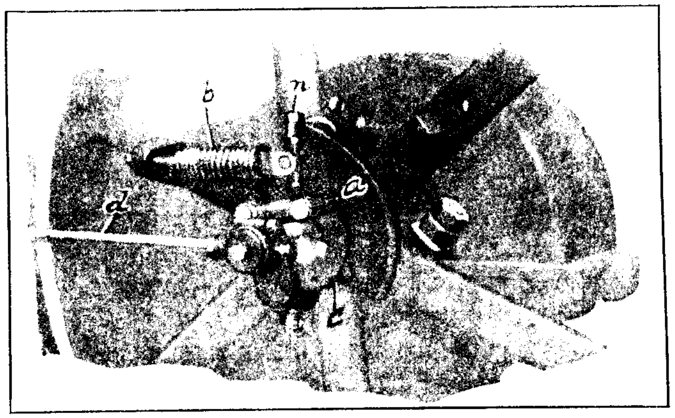


FIG. 3.—Ingersoll-Rand Flywheel Governor, for Class FR-1 Compressors.

The above governor, slightly modified, is used also by the **Pennsylvania Pump and Compressor Co.** and the **Sullivan Mach'y Co.**

Ingersoll-Rand Co., for its class FR-1 steam-driven compressors, employs an automatic-cutoff flywheel regulator or governor, controlled in turn by a flyball governor on the steam cylinder. Fig. 3 shows the flywheel element, consisting of a weight *a*, which, by the pull of spring *b*, releases the part *c* when the compressor speeds up by reason of a decrease in receiver pressure. Then, through rod *d*, the position of the cutoff is altered to reduce speed. Spring *b* is adjustable by hand by screw *n* for a speed change from full to one-half the rated speed of the compressor, while still retaining the automatic cutoff feature throughout the entire range of speed. Turning screw *n* clockwise brings the spring attachment nearer the axis of weight *a*, and reduces the compressor speed; turning it counter-clockwise increases speed. For constant-speed regulation the "RA 39" intake unloader (described later) is used in connection with the flywheel unloader; the result being that the proportion of unloaded time to full-loaded time is minimized.

When working conditions are such that the compressor must operate unloaded much of the total time, but with recurrent full-load demands, a special steam regulator is also used in combination with the above governing devices. This regulator automatically reduces speed to about one-half, when the compressor is running unloaded.

For the Ingersoll-Rand compound two-stage compressors, with piston steam valves, the cutoff valve of the high-pressure cylinder is controlled by the governor shown in Fig. 4. An oil pump *a*, chain-driven from the compressor crank-shaft to sprocket, *s*, delivers oil under pressure to the plunger chamber *c*. The oil pressure forces upward plunger *d*, which carries weight *e*. The vertical movement of *d* is transmitted by rack *g*, pinion *h*, and sprocket *i*, through a chain, to the cutoff valve stem, rotating the stem and thus varying the cutoff. Admission of oil to chamber *c* is controlled by a by-pass valve *f*, which is operated through the pantograph *j*, by the movement of plunger *d*, or of the weight and lever *k*. When the compressor speed (and therefore the speed of the oil pump) produces sufficient oil pressure to raise plunger *d*, valve *f* opens and checks the rise of *d* until the speed increases still further. The maximum and minimum speed screws *l* and *m* are set at the factory to limit the maximum running speed and to prevent such shortening of the cutoff as would stop the compressor.

On starting the compressor, the two halves of the cutoff valve (see Fig. 4, Chapter IV) are brought together until they nearly touch. This gives about a three-quarter cutoff, and since one cylinder port will then always be open the compressor readily starts. As speed increases, the cutoff is shortened to the point of normal running speed. If air consumption ceases, speed is reduced until the compressor just turns its centers. To prevent the speed from exceeding the proper maximum, a safety stop attached to the main throttle (see Fig. 25, Chapter III) automatically shuts off steam.

This governor is also used for duplex compressors with simple steam cylinders, in which case the point of cutoff is changed simultaneously for both cylinders.

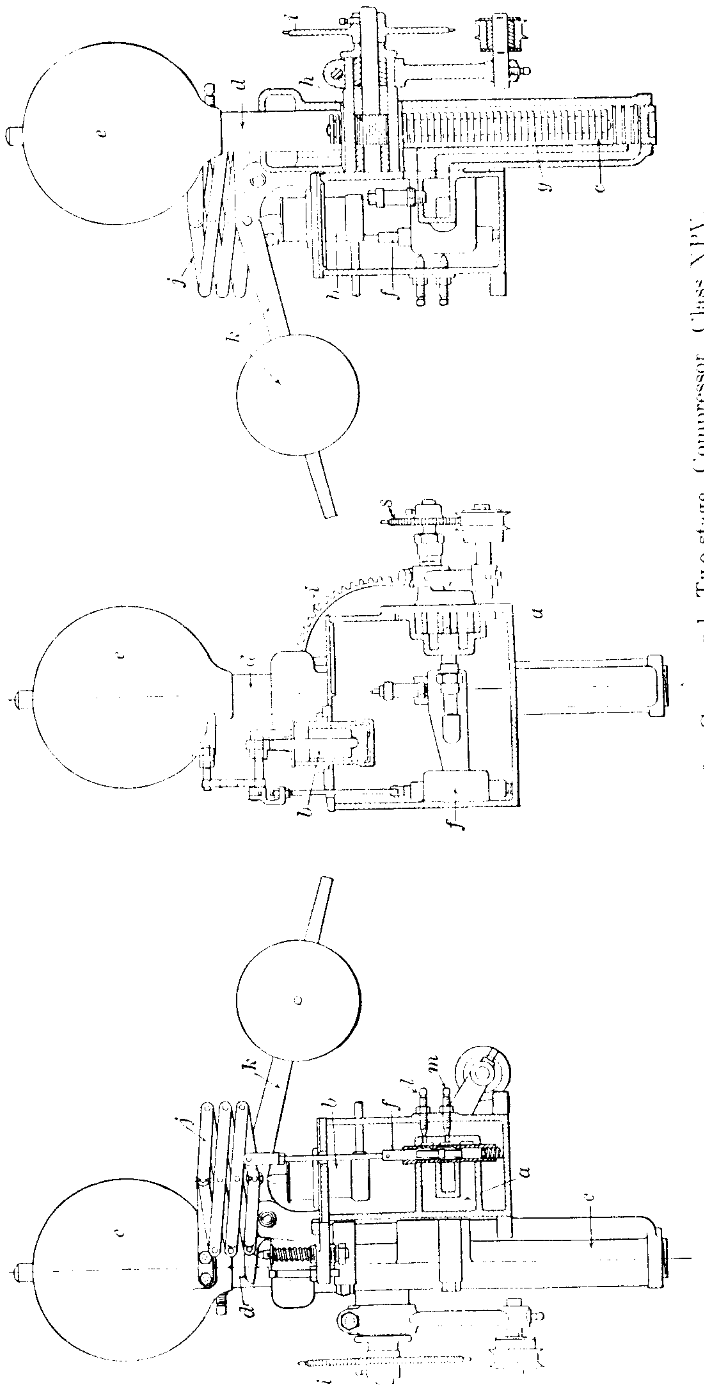


FIG. 4.—Ingersoll-Rand Governor for Compound, Two-stage Compressor, Class XPV.

Norwalk Co.—Fig. 5 shows a governor still to be seen on some of the older steam-driven compressors of this company. It is included here to illustrate a simple type entirely different from the examples already given.

Above a balanced throttle valve in the main steam pipe is a small air cylinder *B*, having a heavy piston. At the side of *B* is a spring-controlled valve, connected by a pipe with the receiver, or with the air delivery pipe leading to it. The spring above this valve is so adjusted that the compressed air will lift the valve and pass through it at any desired pressure. When the receiver pressure exceeds this limit the valve admits air to cylinder *B*, thus raising the heavy piston and partly closing the throttle. To prevent shutting off steam completely, a tapered recess is cut in the piston rod, where it passes through the lower cylinder head (see *R*, in the small cut to left of main figure). As

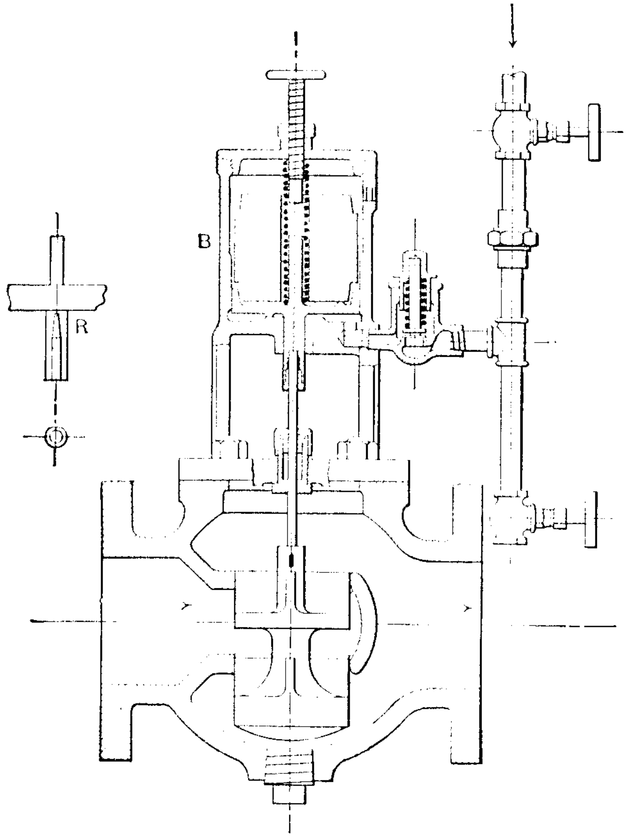


Fig. 5.—Norwalk Governor, Old Design for Steam-driven Compressors.

the piston is forced upward by the air pressure, the varying area of the opening formed by the slotted stem furnishes a graduated escape for the air, and so regulates the piston's movement and through it the amount of opening of the throttle valve. The upward movement of this piston is still further regulated by the screw stop and spring in the top of cylinder *B*. This spring can be so adjusted that, when the piston reaches its highest point, the throttle still admits enough steam to keep the compressor turning its centers.

For their single- and two-stage steam-driven compressors, the Norwalk Company uses a governor of the flyball type, with the attachments necessary for coordinating steam admission to the variations in receiver pressure. This company's belt- and direct motor-driven compressors are provided with intake unloaders of two kinds, as described later.

Sullivan Machinery Co., on its steam-driven compressor WA-6, uses the ball governor shown by Fig. 6, for combined speed and pressure regulation. In general features it resembles the governor in Fig. 2. Compressed air from the receiver enters at *a* to the small cylinder *b*, which contains a spring-controlled valve. When the air pressure exceeds that for which the governor is set, the valve in *b* is opened against the adjustable weight *c*, and through the lever *d* closes the steam throttle valve in *e*. To prevent over speeding, in case of breakage of the governor driving belt on *f*, the spring safety stop *g* closes the throttle through lever *h*.

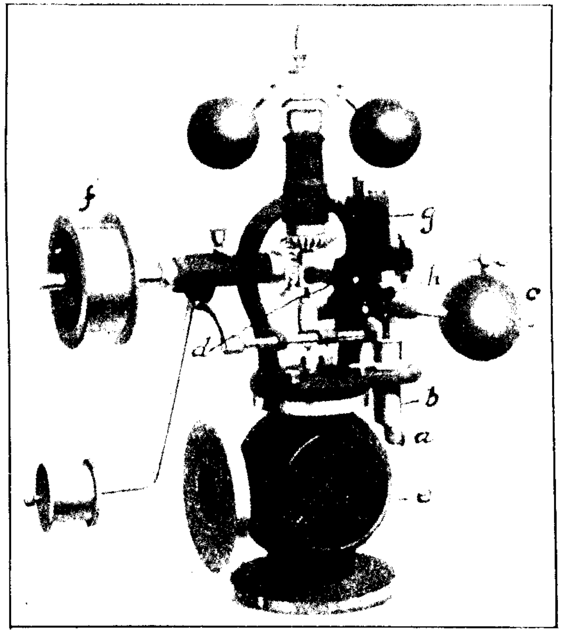


Fig. 6.—Sullivan Machinery Co. Ball Governor for WA-6 Compressor.

Worthington Pump and Machinery Corporation, for its steam-driven compressors of the "Unaflow" type, uses the Jahns automatic governor, made by the Massey Machine Co., Watertown, N. Y. This governor is spring-loaded and directly controls the percentage of cutoff in the steam cylinder. It is operated by bevel gearing from a "lay" shaft, which in turn is driven by gearing from the compressor flywheel shaft.

Fig. 7 is a sectional view of this governor, with the steam valve-motion, these valves being of the poppet type. The governor is mounted alongside of the steam cylinder and compressor frame. The compressor speed may be under control either of the speed element in the governor chamber or of the pressure element shown at the lower left. When the receiver pressure rises above that for which the pressure element is adjusted, the compressor speed is reduced by decreasing the cutoff in the steam cylinder.

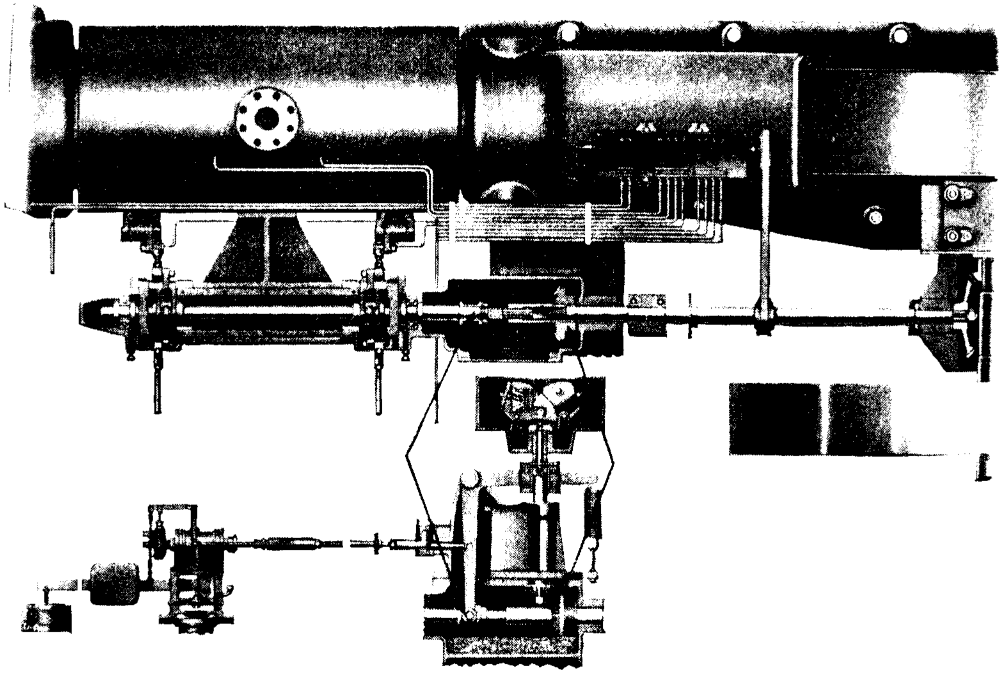


Fig. 7.—Worthington Pump and Machinery Corp. Governor for "Unaflow" Compressors.

As shown in the middle of the illustration, the lay shaft (on the right) and the cam shaft operating the steam valves (on the left) are joined in the governor chamber by a six-point spline, similar to that of the differential gear of an automobile drive shaft. This allows the cam shaft to slide longitudinally when acted on by the governor arm. The latter is caused to move by its connection with the piston rod of the small air cylinder at the lower left, which forms part of the pressure element. The cam shaft is carried in ball bearings, and the cams are tapered so as to modify the point of cutoff of the steam poppet valves, when the shaft is moved endwise. In its operation this valve gear resembles in principle that of an automobile engine, the rotating cams lifting the steam valves by means of hardened steel rollers.

Intake Unloaders for Belt- or Motor-Driven Compressors

Intake or throttling unloaders automatically shut off the intake air of the compressor when the receiver pressure rises above a predetermined point. The compressor then runs unloaded, with small consumption of power, until the air pressure falls to the normal. These unloaders are entirely distinct in principle and operation from the governing devices of steam-driven compressors. Most of them are simple in design. Before discussing unloaders acting on the compressor intake, the unique Holman unloader will be described.

Holman Unloading Valve (Fig. 8). This operates on the discharge side of the compressor. In normal working, the compressed air passes through the retaining valve *a* to the receiver. For the 82-, 134- and 212-cu.ft. Holman compressors, valve *a* is identical with the standard delivery valves.

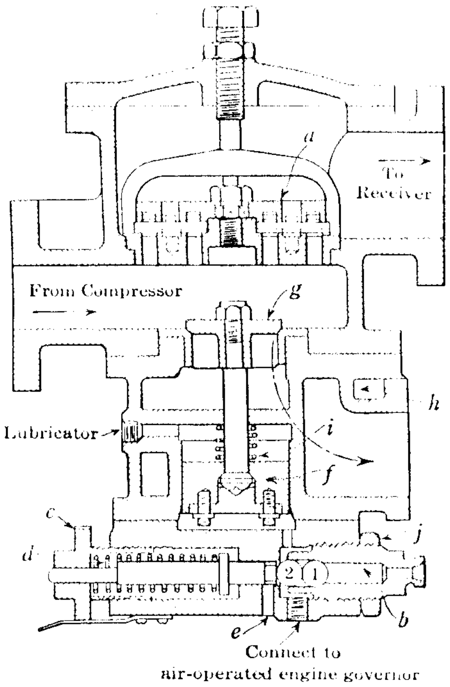


Fig. 8.—Holman Bros., Camborne, England. Unloader for Belt-driven Compressor.

The tube *b* is connected to the receiver by a small pipe. When the receiver pressure rises to the required limit (regulated by cap *c*, acting on spring *d*), the ball *1* lifts and ball *2* closes port *e* to atmosphere. Compressed air then enters below piston *f*, lifting it and valve *g*. The compressed air is now discharged to atmosphere through port *h* in the casing, instead of going to the receiver through valve *a*. When the pressure falls, spring *d* forces the balls back to their former position, closing the passage from the receiver and opening port *e* for the air below piston *f* to escape. Spring *i* then forces down piston *f*, closing valve *g*, and causing the compressed air to pass to the receiver through valve *a*, as before.

To set the balls in tube *b* for normal working, slacken back lock-nut *j*, and screw in tube *b* until both balls are on their seats. Then unscrew *b* one-sixth of a complete turn and tighten lock-nut. Thus, the difference of pressure at which the unloader will function is varied by giving the balls more or less movement. The makers state that the unloader will maintain a nearly constant working pressure from 100 lb. downward—within a range of about 2 lb.

Piston *f* is oiled through the lubricating opening shown, when the compressor has stopped and while the unloader is still warm. If the unloader operates infrequently, cap *c* should occasionally be unscrewed until valve *g* rises, and is then reset for the required pressure.

Fig. 9, in two vertical sections, shows a variation of the above unloader, as used on the Hele-Shaw-Beacham rotary compressor (see also Fig. 51, Chapter III).

When the pressure of the receiver air, entering at *a*, reaches the normal for which the unloader is set, the balls *bb* of the pilot valve rise and admit the compressed air through port *e* to the under side of the hollow unloading-valve piston *d*. On lifting, the unloading valve closes the inlet *cc* to the compressor cylinders, and at the same time lifts auxiliary valve *ff*, the space behind which is connected through *g* with the discharge end of the cylinders. Any air leaking into the cylinders thus passes to the atmosphere, preventing undesirable heating.

On starting the compressor, the unloading valve can be raised by the hand lever *h*, acting on the spindle *i*, so that momentarily no air is compressed.

Ingersoll-Rand "RA-39" Unloader (Fig. 10), for cutting off air at the intake, consists of a balanced disk valve *a*, inserted in the intake pipe and normally held open by spring *g*. When the receiver air, entering at *c*, exceeds the desired pressure, it forces the diaphragm *e* to the right against the spring *f*. The resistances of both springs are adjustable by the screws behind them. Attached to the diaphragm is an auxiliary needle valve *b*, which admits receiver air against the hollow piston *d*, thus closing valve *a*. When receiver pressure falls to the normal, the needle valve closes, the air leaks out from behind piston *d*, and disk valve *a* is forced to the right by spring *g*, again admitting air to the compressor. This unloader, although especially applicable to belt- or motor-driven compressors, may also be used in connection with the flywheel governor for steam-driven compressors. When variable speed is desired, a fixed cutoff with a variable-speed throttling governor is substituted.

A similar unloader is used by the **Pennsylvania Pump and Compressor Co.**

For Ingersoll-Rand class ER-1 compressors, "cylinder relief valves"

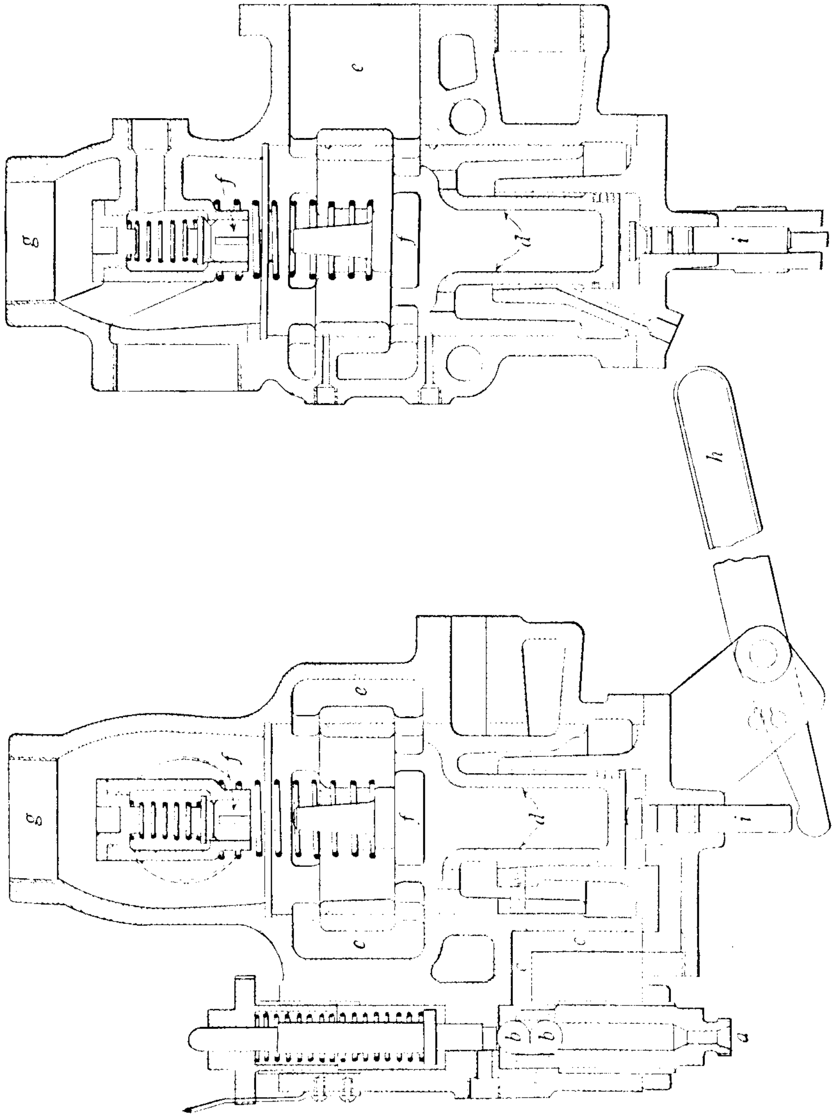


Fig. 9.—Unloader for the Hele-Shaw-Beacham Rotary Compressor (Holman Bros.)

may be used in combination with the RA-39 unloader, to minimize power expended during unloaded periods. The diaphragm and needle valve are omitted, receiver pressure being admitted simultaneously through an auxiliary valve to the unloader and to a relief valve at each end of the cylinder. This shuts off intake air and opens both ends of the cylinder to the atmosphere, thus stopping compression. When the receiver pressure drops below the set limit, compression is resumed.

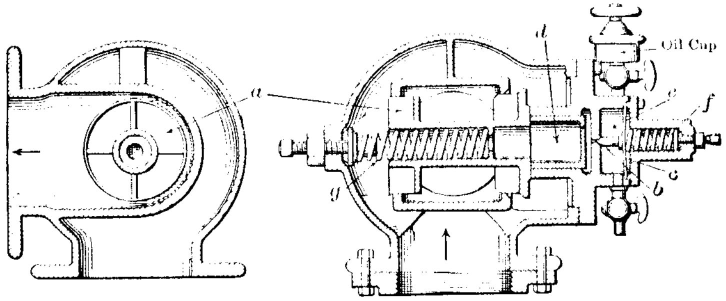


Fig. 10. Ingersoll-Rand "RA 39" Unloader, for Class ER 1 Compressors.

Instead of the relief valves described above, the ER compressors may be provided with the "A 81 regulator," when very close range (say about 2 lb.) is required between the unloading and reloading pressures. This regulator, shown in Fig. 9, Chapter III, consists of a spool-valve, the cylinder of which has two separate connections with the receiver. The lift of the spool tends to raise an adjustable weight suspended from it. When the receiver pressure is sufficient to force the spool and weight upward, live air is admitted to the unloader, which will then hold open the compressor inlet valves. When the receiver pressure is too low to raise the weighted spool valve, air is exhausted from the unloader, thus allowing the inlet valves to operate normally, so that the compressor runs at maximum capacity.

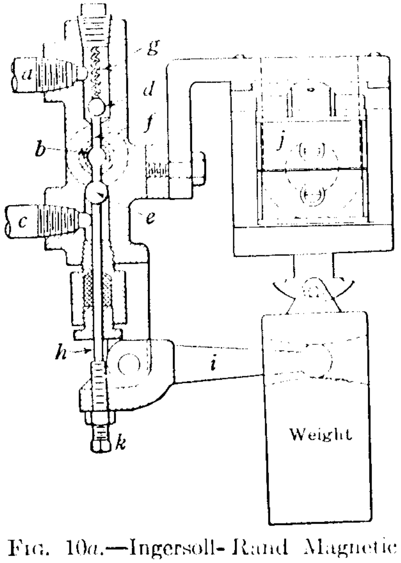


Fig. 10*a*.—Ingersoll-Rand Magnetic Control, A 57.

The Ingersoll-Rand Co. also furnishes a magnetic control (Fig. 10*a*) for its compressor unloaders. A casing containing a pair of ball-valves is connected at *a* with the receiver and at *b* with the unloader, *c* being the exhaust. The ball-valves *d* and *e* are held apart by the spindle *f*, so that only one of them can be seated at any one time. When valve *d* is off its seat, receiver pressure is admitted to the unloader; when valve *e* is unseated, live air is exhausted from the unloader. Spring *g* tends always to keep valve *d* on its seat, but, acting against the spring is the pin *h*, which is forced upward against valve *e* by the weighted lever *i*, suspended from the solenoid *j*, the latter being operated by an automatic control switch from the compressor. Whenever the solenoid is not energized the weight acts to close valve *e* and open *d*, thereby admitting receiver pressure through *b* to the unloader. When the solenoid is energized, the positions of the ball-valves are reversed, thus exhausting through *c* pressure from the unloader. The only adjustment required is made by set-screw *k*, which should be so set that lever *i* is horizontal, when valve *e* is seated; if the weight drops too far when the solenoid is de-energized, it may not pick up on applying the current.

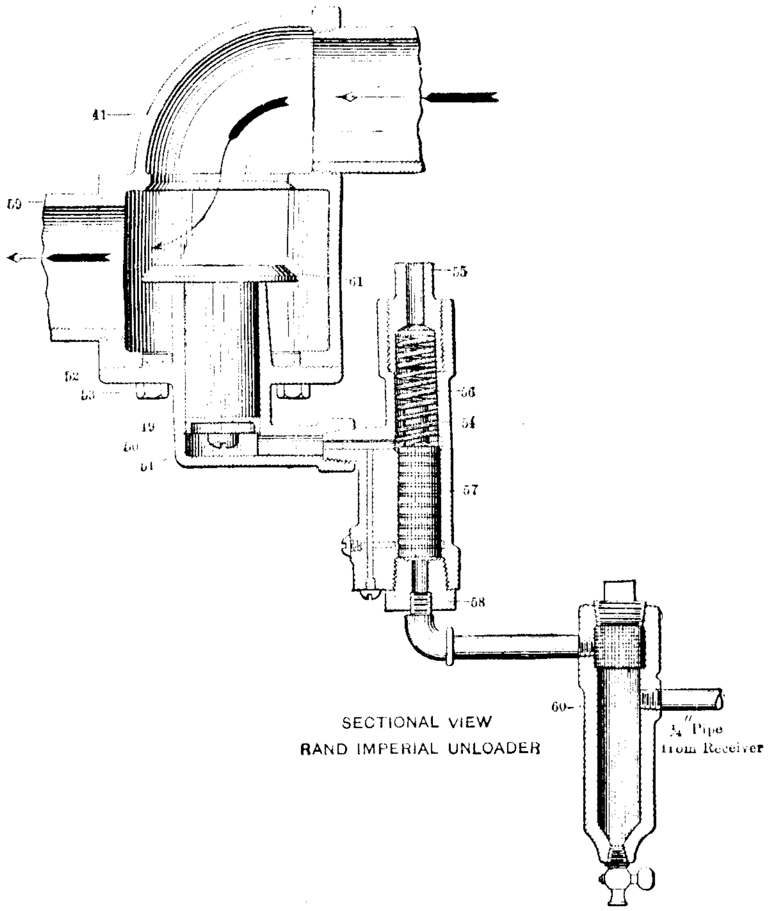


Fig. 11.

Fig. 11, showing an intake unloader formerly used by the Ingersoll-Rand Co., is inserted here to illustrate a type differing materially from the preceding. The intake air enters the cylinder in the direction of the arrows. Chamber (60) is connected to the receiver. As pressure increases, piston (57) moves against spring (56), admitting receiver air through the small port on the lower left of (57) to the under side of plunger valve (61). On reaching its seat this valve closes the intake. Spring (56) is adjusted by the screw-plug (55) for any required pressure. When the receiver pressure falls, on increased consumption of air, the spring forces down piston (57). This closes the lower small air port, and connects the upper horizontal port with the open screw-plug (55).

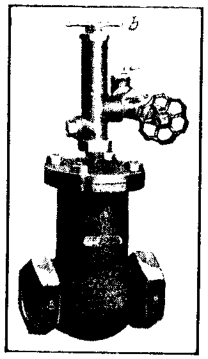


Fig. 12.—Intake Unloading Valve, Norwalk Co.

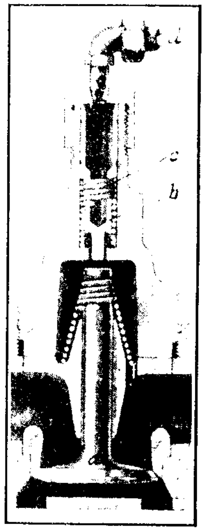


Fig. 13. Poppet-valve Unloader, Norwalk Co.

The air below valve (61) is thus exhausted, causing the latter to reopen the intake passage, and the compressor resumes useful work.

Norwalk Co., for all single-stage belt- or motor-driven compressors, supplies a throttling unloader of simple design (Fig. 12). Attached to the compressor intake is a shut-off valve, operated by a pilot needle valve connected to the discharge pipe or air receiver. Compressed air enters at *a* to the pilot valve, which is adjusted for any required pressure by screw-plug *b*; *c* being the shut-off valve.

The same type of unloader is used for similar service by the **American Air Compressor Works.**

The Norwalk Co. also furnishes to order a poppet-valve unloader, for both high and low pressure (Fig. 13). This is wholly different in design from the preceding. Above the stem of one of the inlet poppets *a* is a small plunger valve *b*, normally held against an upper seat by the spring *c*. In the pipe to the receiver is an adjustable control valve (not shown), set for the desired air pressure. When this pressure is exceeded, the control valve admits compressed air at *d* to the plunger valve *b*, which is forced down to its lower seat and to contact with the stem of the poppet valve, thus holding the latter open until the air pressure falls. While the poppet remains open, the inlet air passes in and out of the cylinder, without being compressed.

This form of unloader makes it unnecessary, when starting the compressor, to "bleed" the cylinders; that is, to discharge from them their content of compressed air. The cylinder of each stage remains unloaded until full discharge pressure has been reached in the preceding stage.

Nordberg Unloader, an elaborate mechanism for motor-driven compressors having Corliss air valves, closes the inlet valve before the end of the forward stroke, which is equivalent to varying the working length of stroke.

In Fig. 14, wrist-plate *w* is driven by rod *a* from an eccentric on the flywheel shaft; another eccentric operates the releasing mechanism through rod *b*, which oscillates arm *c* about the fixed center *d*. On the lower end of *c* is a 3-armed rocker. Arm *i* is linked by rod *j* to the radius fork *k*, which is connected to the pressure governor *l*. Arms *g* and *h* through rods *e* and *f*, operate the releasing cams *n* and *o*, on the inlet-valve spindles. When the air consumption is normal, rocker arms *g* and *h* remain vertical, under the action of rod *b*, and impart equal movement to both cams. When the receiver pressure increases, arm *i* moves upward, arms *g* and *h* take an inclined position and rods *e* and *f* alter the point of release of the valves.

Fig 15 shows the releasing mechanism. Mounted on the valve spindle is the 3-armed rocker. The wrist-plate link is connected to arm *a*, the releasing latch *d* to arm *b*, and the governor cam-arm *e* to arm *c*. Since *e* is connected also to the governor by rod *f*, it has a compound motion; it swings from its swivel pin at the top, its position being adjusted laterally by the governor's action. The cam slot has two circular arcs, struck from the center at upper end of *e*. As the roller on arm *g* moves along the cam groove, latch *d* is alternately released and engaged, when the roller passes the jog in the cam.

Figs. 16, 17 and 18 are indicator cards from a two-stage compressor, each cylinder having its own governor. The upper cards are from the intake, the lower from the high-pressure cylinder. Referring to the intake card of Fig. 17, the inlet valve remains open to about mid-stroke *b*; the releasing gear then acts and the valve closes. From *b* to end of stroke, the air expands below atmospheric pressure. On the return

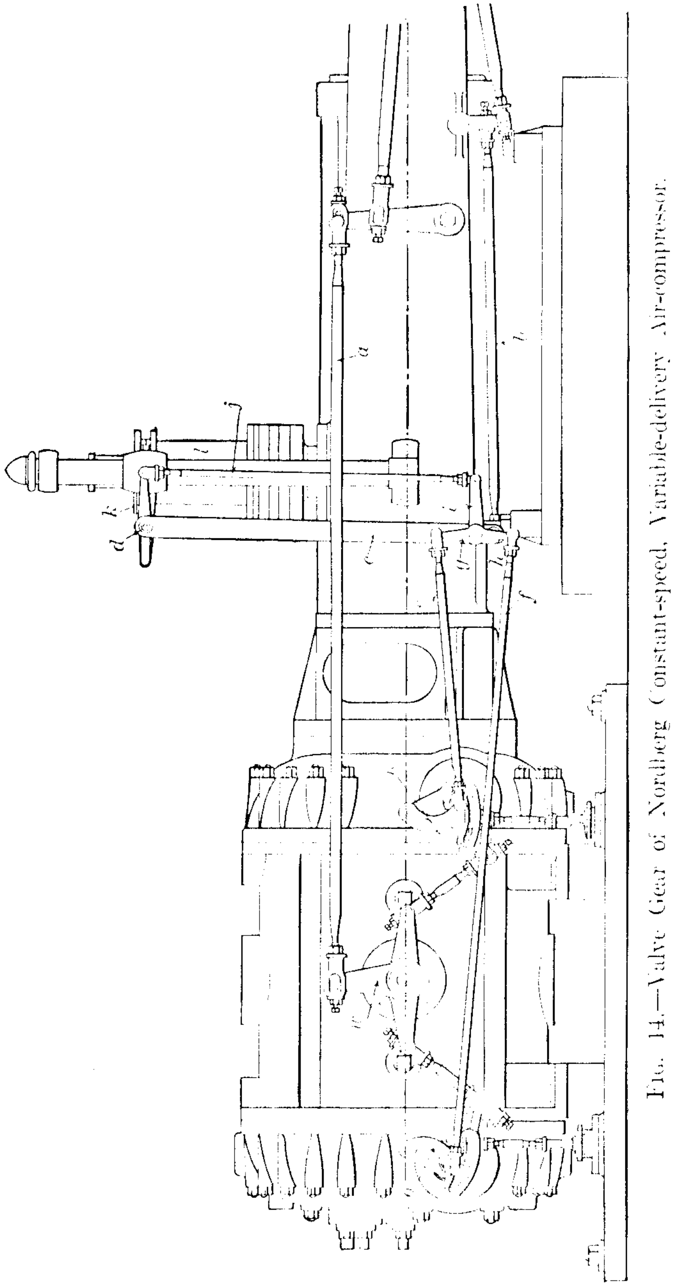


Fig. 14.—Valve Gear of Nordberg Constant-speed, Variable-delivery Air-compressor.

stroke, the compression line nearly coincides with the expansion line from *c*, until atmospheric pressure is reached at *b*, after which compression proceeds as usual. The inlet valves of the high-pressure cylinder act similarly, except that the expansion and re-compression of the air is from receiver pressure, instead of atmospheric pressure. In Fig. 18 the cards show the small amount of work done when the compressor is under nearly zero load.

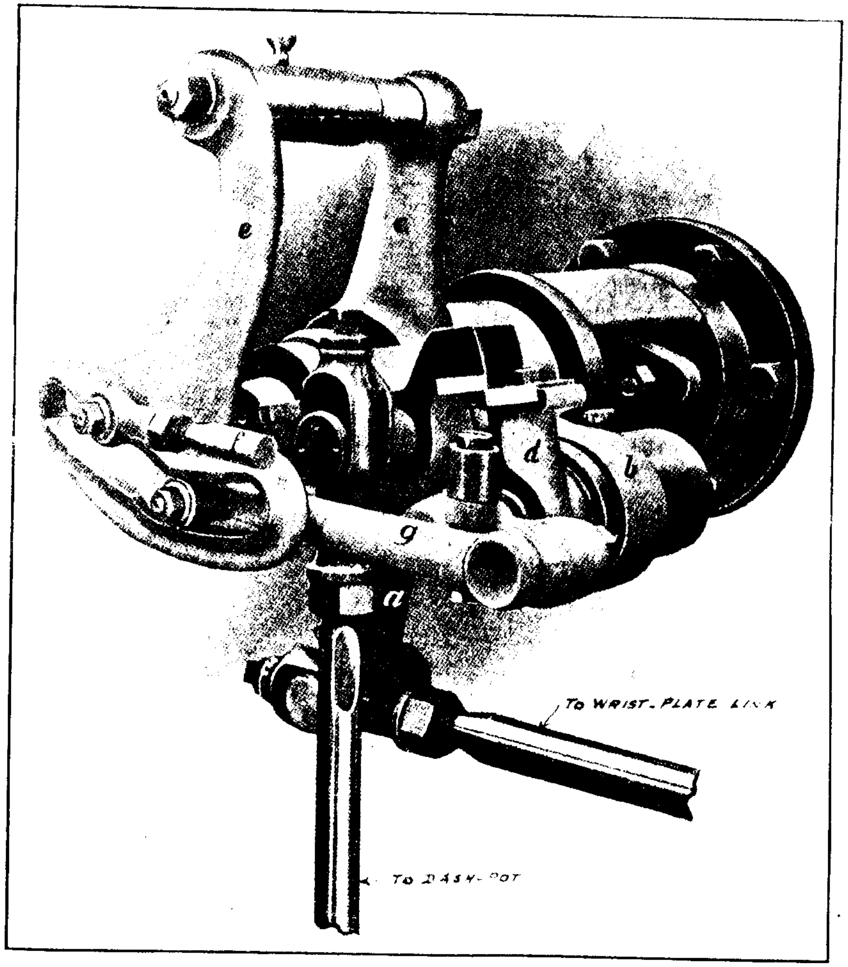


Fig. 15.—Detail of Valve Gear Shown in Fig. 14.

Sullivan Machinery Co., for its angle-compound, motor-driven compressors, uses an intake unloader similar in principle to the Ingersoll-Rand RA-39 (Fig. 10). It is controlled by a pilot valve, adjustable for any desired air pressure.

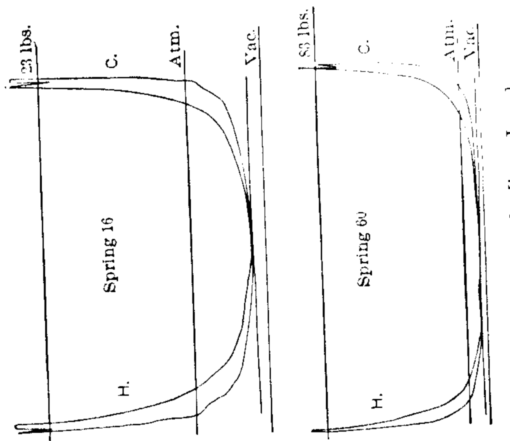


Fig. 18.—Nearly Zero Load.

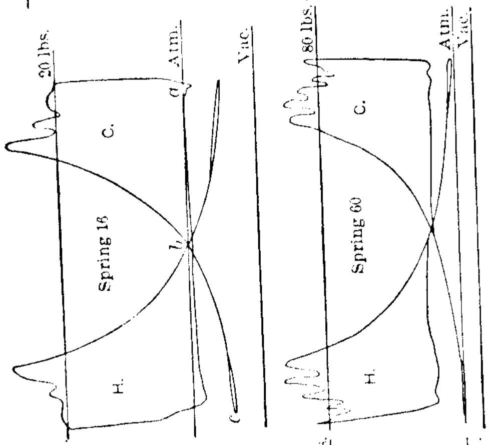


Fig. 17.—Half Load.

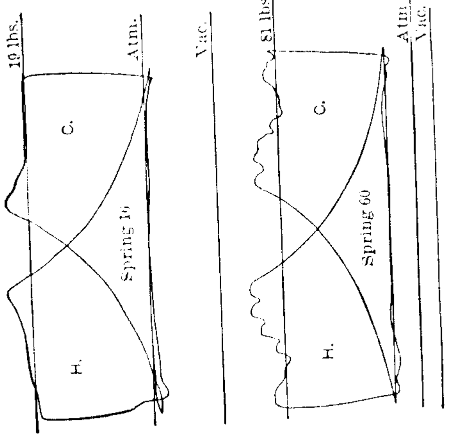


Fig. 16.—Nearly Full Load.

Coordinated with the unloader, but operating independently of it, is a "relief valve." When the unloader has been acting for a few revolutions of the compressor, and the intercooler pumped out by the high-pressure cylinder, the relief valve exhausts any air that may have leaked back into the compressor, thus preventing an undue rise of cylinder temperature.

SELECTIVE OR STEP CONTROL FOR BELT- AND MOTOR-DRIVEN COMPRESSORS

These have been developed for the most part during the past ten years. They exercise a step or progressive control over the operation of one or more of the inlet valves, their effect varying in degree according to the receiver pressure. In some cases compressors are provided with both intake unloaders and step controls, a combination which affords the most efficient form of regulation for minimizing the consumption of power.

As a rule, two- or three-step control is applied to the smaller sizes of compressor; five-step control to the larger sizes.

Chicago Pneumatic Tool Co. Its two-stage power-driven compressors are provided with 2-, 3-, or 5-step regulation, produced by holding open automatically by an unloader certain especially designed inlet valves, called "unloading inlet valves," thus returning to the atmosphere part of the air entering the cylinder. The design of these valves is similar to that shown by Fig. 23 of the Norwalk Co.

In *two-step regulation*, when one set of unloading inlet valves on each cylinder are held open, the load is reduced by 50 per cent; when both sets on both cylinders are held open, the load is entirely removed.

Fig. 19 shows the application of *three-step regulation*. The unloading inlet valves are held away from their seats by receiver pressure under control of two unloaders. These valves thus become inoperative, permitting atmospheric air to pass freely in and out of the cylinder, without being compressed. Fig. 20 is a vertical section through the unloader, two of which are shown in place in the system at *a* and *b*, Fig. 19. In Fig. 20, the valve plate *c* is held normally against its upper seat by the spiral spring, which is adjusted by the nut *d* for any required air pressure. When this pressure is exceeded, valve *c* leaves its upper seat; and, as a larger area of the valve is thus exposed to receiver pressure, the valve is instantly forced against its lower seat. Receiver air then passes through the ball check valve *e*, and thence through *f* to the compressor unloading inlet valves, which are thus held open and inoperative, as stated above.

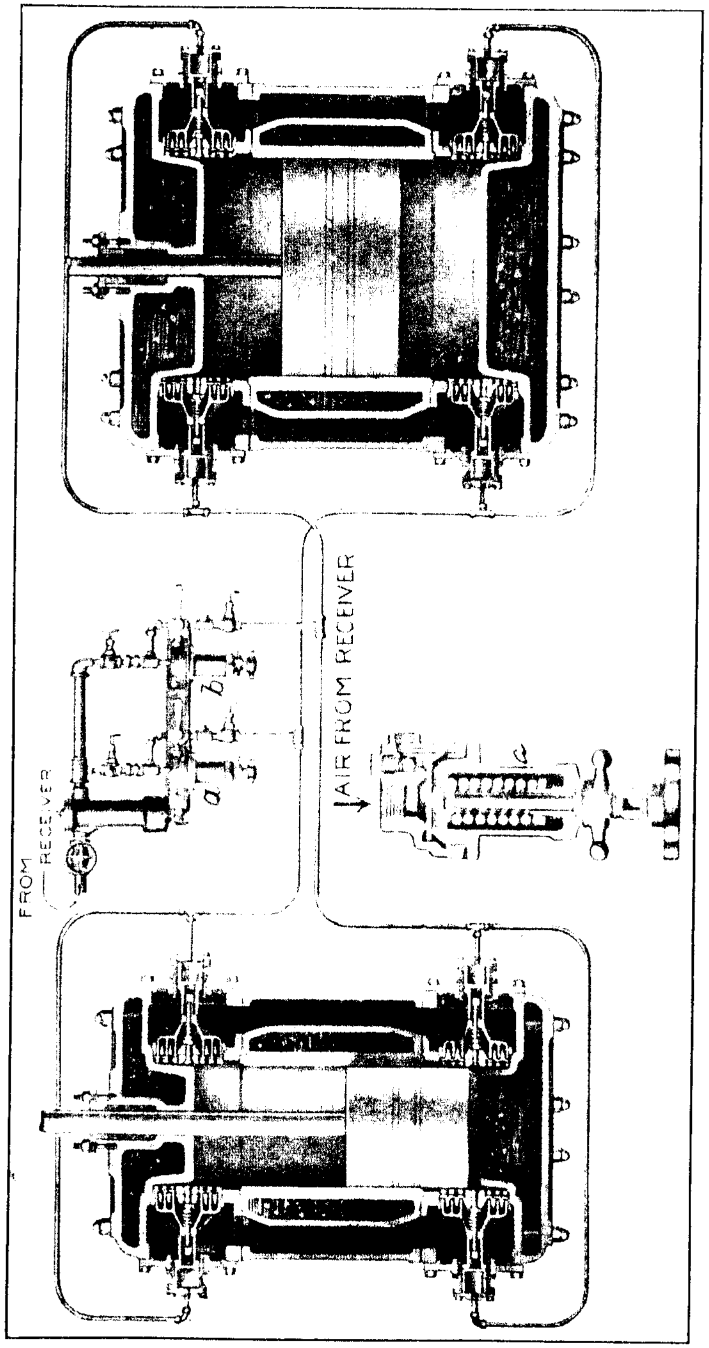


Fig. 19.—Three-step Capacity Regulation for Two-stage Compressors. Chicago Pneumatic Tool Co.

For full compressor capacity, the inlet valves designed for unloading function normally. For half capacity, one of the two unloaders operates, holding open the unloading inlet valves of the head end of one cylinder and the crank end of the other. For zero capacity, or complete unloading, the second unloader acts, holding open the rest of the unloading inlet valves.

With *five-step regulation*, the compressor is similarly caused to operate at full, three-quarter, half, one-quarter and zero capacity.

Ingersoll-Rand Co.'s five-step clearance control differs materially from the preceding. A separate unloader is not required, its functions being performed by small pilot valves mounted above the unloading inlet valves.

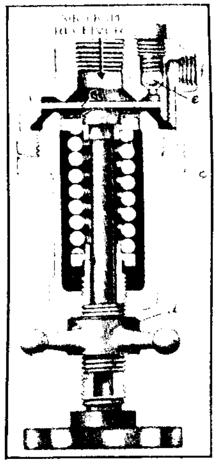


Fig. 20.—"Simplate" Unloader, Shown in Place in Fig. 19, at *a* and *b*. Chicago Pneumatic Tool Co.

Fig. 21 shows the air cylinder, each end of which has two "clearance pockets" *b*, *b*, immediately under the unloading inlet valves, each of these having its pilot valve *a*. For a two-stage compressor, the ratio between the volume of the clearance pockets in the high- and low-pres-

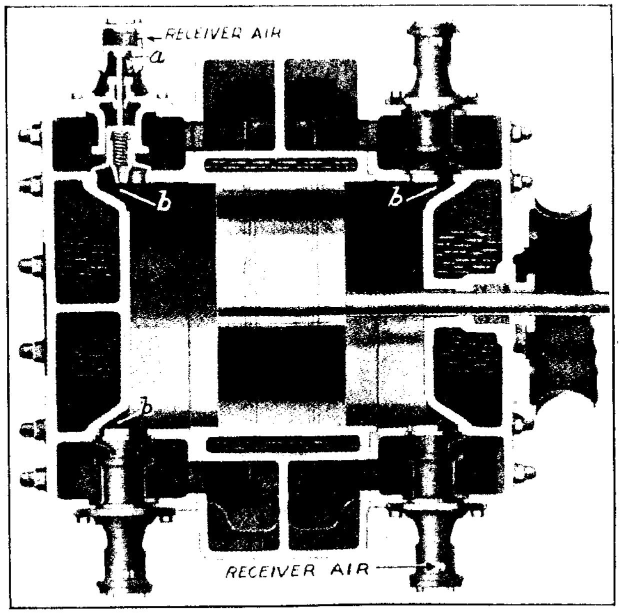
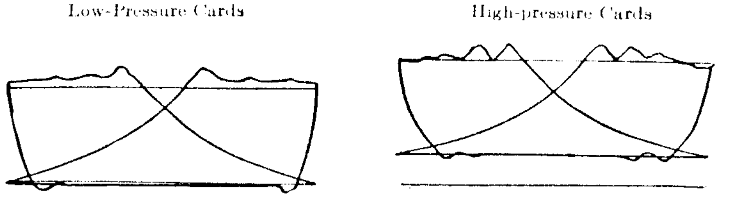
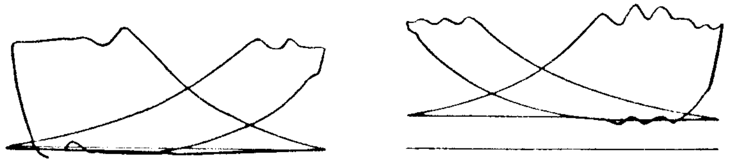


Fig. 21.—Ingersoll Rand Co. Air Cylinder of Direct Motor-driven Compressor, Showing the Clearance Pockets, with Their Unloading Valves for 5-step Control.

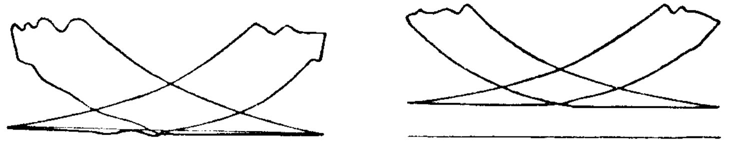
sure cylinders is the same as the ratio of the cylinder volumes (see Chapter VI). For a single-stage compressor, with two cylinders, the pockets of both cylinders have the same volume. The action of the unloading inlet valves is controlled by admitting receiver air above the pilot valves, the valves being arranged to open in proper sequence. If when the compressor is running at full load, the demand for air decreases, the receiver air forces down the pilot valve of one clearance pocket in each cylinder. This opens the corresponding inlet valve and reduces the volumetric capacity of the compressor to approximately 75 per cent of normal. If the demand for air continues to decrease, the second set of clearance pockets are opened, reducing the capacity to 50 per cent, and so on; until, when all the pockets are opened the compressor is completely unloaded.



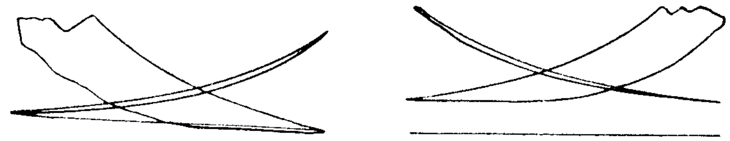
Full Load—100% Capacity; 100% Power. Indicated H. P., 419, for Measured Output of 2338 cu. ft. per min.



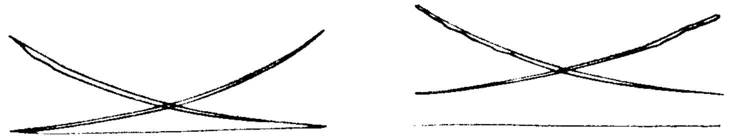
Three-quarter Load—75% Capacity, 77.5% Power. I.H.P., 325 for 1753 cu. ft. per min.



One-half Load—50% Capacity, 55% Power. I.H.P., 230 for 1169 cu. ft. per min.



One-quarter Load—25% Capacity, 30% Power. I.H.P., 126 for 584 cu. ft. per min.



No Load—Zero Capacity, 3.6% Power. I.H.P., 15 for no cu. ft. per min.

Fig. 22.—Indicator Cards Showing Results of 5-step Clearance Control. Ingersoll Rand Co.

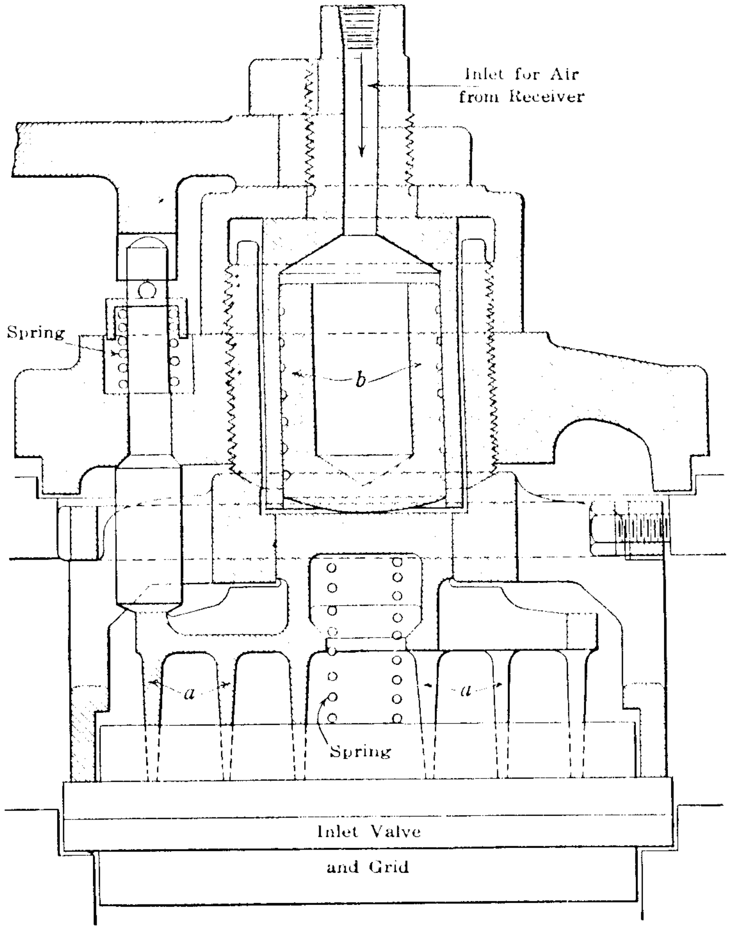


Fig. 23.—Norwalk Co.'s Proportional Unloader.

The effect of this regulation is exhibited by the indicator cards in Fig. 22. In each case the air is compressed to full terminal pressure, the volumetric capacity only being altered. The cards also show that the ratio of the compressor output to the power input is nearly constant.

To adapt the class "PRE" compressor to the starting characteristics of the synchronous motor used to drive it, each cylinder has a relief valve, which is opened during the starting period, so that the discharge air is temporarily by-passed to the atmosphere.

For the "PO" type of two-stage compressor, direct-driven by oil engine (see Chapter III), the unloader is similar to the above, but is differently arranged. When it operates, the air is by-passed back to the intake passage of the low-pressure cylinder.

Norwalk Co.'s three-step "proportional unloader" for two-stage compressors is shown by Fig. 23. It consists of a set of fingers *a*, operated by a plunger *b*. The fingers force down the rings composing the inlet valve, and hold them open, when the desired maximum air pressure is reached. This action of the plunger and fingers is caused by the operation of a pair of pilot valves. Compressed air from the receiver is transmitted to the pilot valves, which are set for the predetermined pressure. From the pilot valves small pipes lead to the unloading inlet valves of the rear ends of the low- and high-pressure cylinders, with branch pipes to similar valves on the front ends of the cylinders. The arrangement of the pilot valves and piping is shown by Fig. 24.

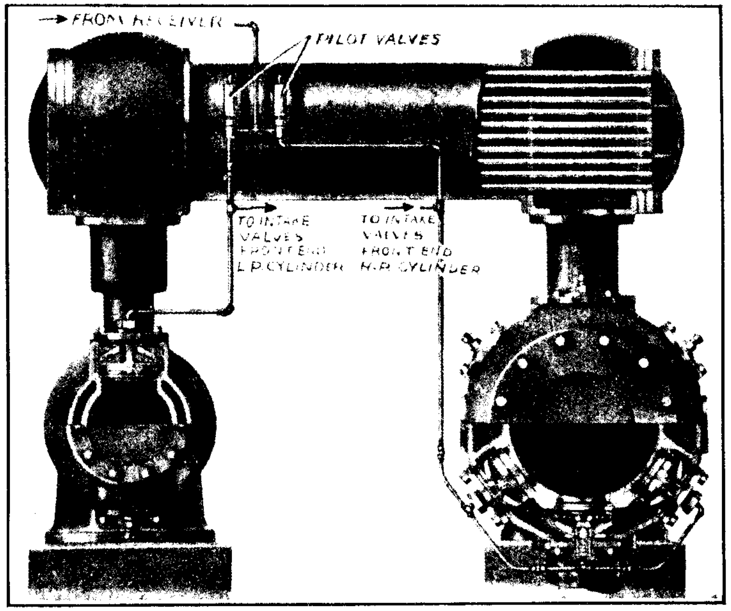

FROM RECEIVER
PILOT VALVES
TO INTAKE VALVES FRONT END L.P. CYLINDER
TO INTAKE VALVES FRONT END H.P. CYLINDER

Fig. 24.—Arrangement of Pilot Valves and Piping for the Norwalk Proportional Unloader.

The timing of the opening of the unloading valves of the two cylinders is such that the mechanical balance of the compressor is preserved. By this device the compressor is unloaded in three steps: full, half and no load, the power required being proportional to the quantity of compressed air delivered.

Worthington Pump & Mach'y Co.'s unloader, for one-, three- and five-step control, is similar in principle to the Norwalk, though the plunger and fingers are differently designed, to adapt them to the "feather" valves used in the Worthington compressors (see Chapter VII). Fig. 25 shows one of the unloading inlet valves, which are set in the upper part of the air cylinder. The pilot or trigger valve for operating the unloading valves is shown by Fig. 26. It is connected to a pipe from the receiver.

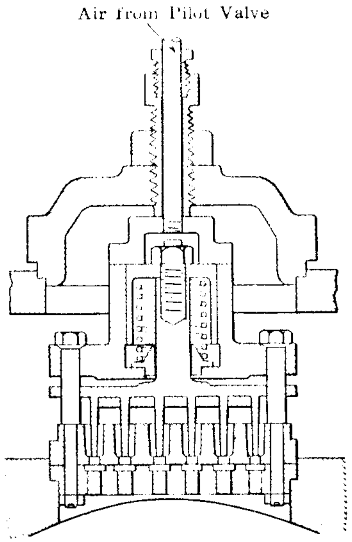


Fig. 25.—Unloading Inlet Valve, Worthington Pump and Machinery Corporation.

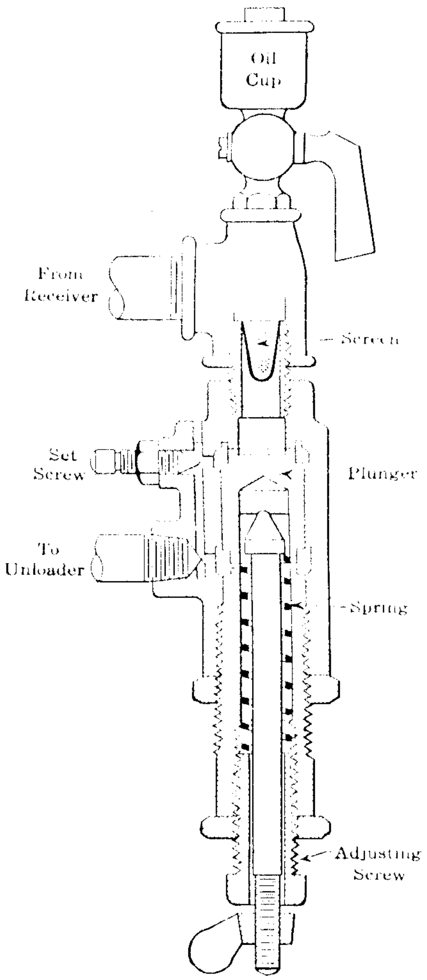


Fig. 26.—Worthington Trigger Valve for Unloader.

Governors for Turbo-Compressors

As turbo-compressors are infrequently used for the kinds of service covered by this book, a brief general statement only will be made here regarding the complicated mechanism of their governors.

The usual mode of regulation is to throttle the air at the inlet. For this, a valve of the butterfly type is commonly used, together with a relief valve and a self-acting non-return valve in the discharge pipe. When the load on the compressor drops to say 65 or 70 per cent of normal, the intake butterfly valve begins to close, its throttling action being progressive as the air demand continues to fall. At zero air consumption, the non-return and butterfly valves close entirely, and the relief valve opens. The compressor then operates in vacuum, consuming but little power. When the air demand is resumed, causing a pressure drop in the discharge pipe, the butterfly valve opens, the relief valve closes, and the compressor again picks up its pressure.

Governors of the above general type are employed, for example, by Adamson & Co., Fraser & Chalmers (both British), and the General Electric Co., New York. The inlet control of the last-named maker is a very elaborate mechanism, being interconnected with a special centrifugal governor on the power end of the compressor. For its steam-turbine-driven turbos, the Ingersoll-Rand Co. employs an adaptation of the flyball governor, in which spring-loaded weights, operated by a system of levers connected to a sliding sleeve on the governor spindle, are substituted for the flyballs. It is driven by worm gearing from the main compressor shaft, and regulates steam admission according to the variations in air demand.

These governing devices for turbo-compressors are extremely sensitive. They are capable of maintaining a nearly constant pressure under wide variations in the volume of air consumed (see Turbo-compressors, latter part of Chapter III).

CHAPTER IX

AIR RECEIVERS AND AFTERCOOLERS

In its common form the receiver consists of a cylindrical shell of steel plate, resembling a steam boiler with or without tubes. It has pipe connections to the compressor and air main, a pressure gage, safety-valve, drain cock, and manhole. The vertical form, Fig. 1, is often preferable, as it occupies less floor space. Fig. 2 shows a horizontal, tubular receiver.

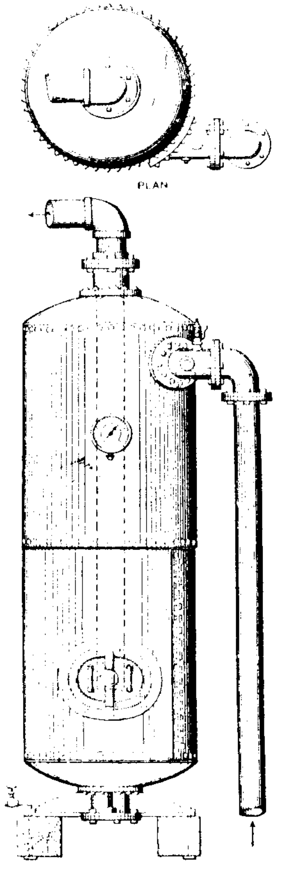


Fig. 1.—Vertical Receiver

Receivers are usually built to stand a test of 165 lb. cold-water pressure, for working under pressure of 100 120 lb. Except for charging compressed-air locomotives, higher pressures than these are rarely necessary for mine service. The shells are single-riveted on circular seams and the larger sizes double-riveted on longitudinal seams; the heads being dished. For best results, the receiver should be placed close to the compressor, or in any case not more than 40 50 ft. distant.

Principal Functions of Air Receivers.—(1) To eliminate the pulsating effect of the strokes of the compressor piston and thus prevent rapid fluctuations of air pressure; (2) to minimize the frictional loss attending the flow of air through the air main and lines of piping; (3) to serve in some degree as an equalizer and reservoir of power; (4) to cool the air before it passes into the main, thus causing it to deposit a part of its moisture in the receiver, whence it is drained off.

Regarding the first point, the volume of the receiver should be large enough in proportion to that of the compressor cylinder to prevent any material rise of receiver pressure by the volume of air forced into it at each stroke. If the compressed air passed directly into the main, large fluctuations of line pressure would occur. During the periods of acceleration of flow, the frictional resistance in the pipe would be increased, and at the end of each stroke the compressor piston would have to force the air out of the cylinder against a pressure momentarily greater than the normal.

The second function of the receiver is best fulfilled by placing an auxiliary receiver near the point where the compressed air is used. Just as the receiver at the compressor diminishes the momentary rise of pressure in the main due to each stroke of the piston, so a second receiver close to the machine using the air reduces the drop of pressure as each cylinderful of air is drawn off. The two receivers maintain a nearly constant flow of air through the main connecting them, thus minimizing friction and loss of pressure.*

For mine service the second receiver would be placed underground. For air drills only, underground receivers are not often used, but they are a necessity for compressed air pumps and hoists. They also permit a further deposition of moisture, thus rendering the air dryer and more suitable for expansive-working engines. To accomplish this most effectually, the second receiver should be placed at the point in the pipe line where the air reaches its lowest average temperature. Underground receivers sometimes consist of a chamber excavated in the rock, with cemented or asphalted walls. The mouth of the chamber is closed by a concrete dam of two parallel walls, with a layer of cement grout between them. In the dam are set a manhole, flange connections for the inlet and outlet pipes, and a drain pipe and cock close to the floor. The latter is opened from time to time, to blow out the accumulated water and sediment. Such receivers are of relatively large volume, and cost much less than ordinary steel-plate receivers of equal capacity. For details of design of an underground receiver, see *Engineering and Mining Journal*, Jan. 31, 1914, p. 275.

The third function of the receiver is apt to be exaggerated. To act to any great extent as a reservoir of power the receiver must be very large. For example, take a 20-in. compressor, working at 60 lb. pressure. To meet the corresponding air demand for only 1 minute after the compressor is stopped, and not have the pressure fall more than 15 lb., the receiver volume would be nearly 10,000 cu.ft., or say 5 ft. in diameter by 50 ft. long. Again, if the compressor were running at a constant speed and the demand for air should suddenly increase 25 per cent, as

---

* The flow of air in pipes, and its frictional losses are treated in Chapter XV.

might happen in starting several more machine drills, a receiver of the above size could meet the extra demand for only 4 minutes. Long mains of large diameter assist in equalizing the flow of air, but it is obviously cheaper to employ piping of moderate size, in connection with a receiver of ample volume.

The fourth function of the receiver is important. Considerable moisture is always present in compressed air, due to the natural humidity of the atmosphere, especially in warm weather. The velocity of flow of the air discharged by the compressor is greatly reduced on entering the receiver; and on cooling the air deposits part of its moisture, which otherwise would be conveyed into the piping, and thence to the machines using the air. Moisture tends to wash away the cylinder lubricant, and so increase wear and consequent leakage of air past the piston. This is especially true of high-speed machines, as drills and small air hoists, in which the wearing surfaces are of small area. Moisture collecting in pipe lines may also cause "water hammer," reduce the air passage by accumulating at low points, and in winter may freeze and burst the pipes. Wet air, freezing in drills, etc., clogs the exhaust and increases back pressure. The receiver should be large enough to drain the air thoroughly. In the ordinary sizes of receiver the results are often quite imperfect, because the air passes through too rapidly to permit any considerable drop in temperature. The inlet and outlet pipes of the receiver should be placed in proper relative positions. If they point toward each other, the strong through current causes the air to pass out before it has cooled sufficiently to drop much of its entrained moisture. One mode of connecting the pipes is to place the inlet on one side, near the end of the receiver, while the outlet is at the opposite end, in the middle of the head. The air is thus forced to change its direction of flow. Or, as in Fig. 1, both pipes may be connected near the top, the outlet pipe being carried nearly to the bottom, where the air is likely to be slightly cooler (and therefore dryer) than at the top. As the inlet pipe shown in this case is connected tangentially to the periphery of the receiver, a rotary motion is imparted to the body of air, so that each particle remains longer under the cooling influence of the receiver. Due to the reduction in volume of the air caused by cooling, the discharge pipe is of smaller diameter than the inlet.

Receiver Aftercoolers (Figs. 2 and 3) differ from the ordinary shell receiver in being provided with baffle-plates and with closely spaced iron, copper, or aluminum tubes, through which cold water is circulated. A direct, through current of air is prevented by the baffles, and a more thorough cooling is effected by the passage of the air in thin streams between the water-cooled tubes. The shell is of steel plate, with inner, cast-iron heads, in which the tubes are free to move longitudinally as they expand or contract. The arrows in Fig. 2 show the direction of flow of air and water. A good aftercooler will lower the temperature of the compressed air to within say 15° or 20° of that of the water.

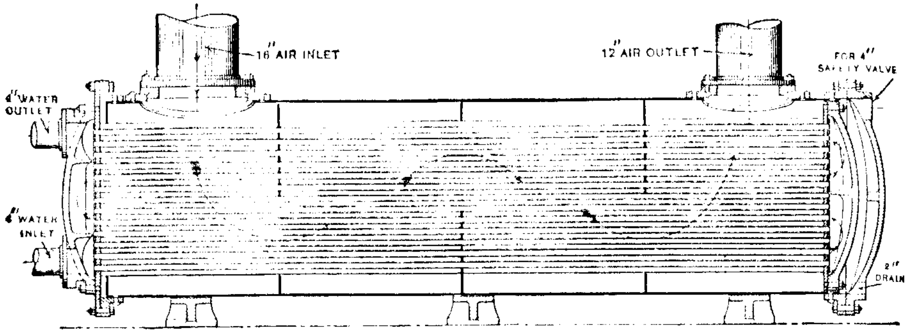


Fig. 2.—Horizontal Receiver-aftercooler.

Fig. 3 shows the latest type of Ingersoll-Rand aftercooler. The air enters at *a*, flows upward through the first shell, thence into the top of the second shell and downward to the discharge at *b*. Before discharge, the air passes over a moisture separator in the tee *c*. This consists of a number of "scrubber" bars or ribs, assembled in herringbone form; these catch the moisture as the air current impinges on them, and cause the water to deposit in the bottom of *c*, whence it is drawn off from time to time through a valve. If desired, this aftercooler may be provided with cast-iron pedestals, for standing horizontally.

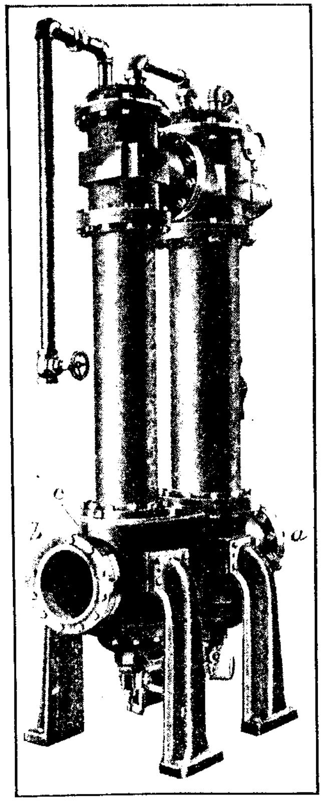


Fig. 3.—Ingersoll-Rand Class VM-2 Aftercooler.

Due to the small storage capacity of the form of aftercooler shown in Fig. 3, the air from it should pass to an ordinary receiver before going to the pipe-line.

The type of receiver aftercooler shown by Fig. 2 is similar in construction to the intercooler used for stage compressors (Chapter VI). When of ample size and placed close to the compressor, it is much more efficient in economizing power than the plain, tubeless receiver. This is true because it reduces more completely the temporary rise in pressure due to the heat of compression, against which the piston acts when forcing the compressed air out of the cylinder (Chapter II, equations 17 and 18). As most of the heat of compression is lost before the air is used, this saving of power is worth while, since it is accomplished without appreciable operating cost.

Volume of Cooling Water for Aftercoolers

Gallons per 100 cu.ft. free air, for different water temperatures (Ingersoll-Rand Co.)

Water temperatures	60°	70°	80°	90°
For 80–100 lb., single-stage compression . . .	4.0	4.5	5.2	6.0
For 80–100 lb., two-stage compression	2.5	3.0	3.5	4.0

Ante-coolers, sometimes used to cool the compressor intake air, are especially desirable for large plants, or, for compressors operating in hot climates. They are similar in design to aftercoolers, their effect being to reduce the temperature and volume of the air entering the compressor cylinders and therefore increase the volumetric capacity of the cylinders, in terms of weight of air compressed per stroke. The effect of temperature of the intake air upon the thermodynamic operation of a compressor is discussed under "Arrangements for Admitting Intake Air," Chapter VII.

Volumetric Capacity of Receivers.—Referring to the preceding statements regarding the third function of receivers, the capacity should be sufficient to prevent rapid or great fluctuations of air pressure, and must therefore depend upon the type of compressor and the kind of service. The volume of discharge per stroke from compressor to receiver varies with the size of the cylinders and the ratio of compression. It should be remembered that the total discharge from the cylinder takes place during one-quarter to one-half the length of stroke, according to the pressure produced and whether the compressor is single-stage or two-stage.

A rough rule for ordinary rock-drill service is to allow a receiver capacity of 100 cu.ft. per 800 to 1,000 cu.ft. of free air compressed per minute. For stationary, constant-running air engines, like pumps, the capacity may be smaller.

For a rational determination of receiver capacity the formula expressing Boyle's law (Chapter II) may be used, $V' = \frac{VP}{P'}$; where V' =

required capacity, cubic feet; V = compressor piston displacement, cubic feet per minute; P = initial air pressure, pounds; P' = working gage pressure, pounds. At sea-level, P = 14.7 lb. (a table of atmospheric pressures at different altitudes is given in Chapter XII). For use at sea-level, the formula becomes $V' = \frac{V \times 14.7}{P' + 14.7}$.

Example. If V = 800 and P' = 80, V' = 121 cu.ft. This is the minimum volume necessary.

In view of the previous discussion of the functions of the receiver, a greater capacity than that computed by the above formula is always desirable, to secure economy in the production and use of compressed air. Compressor builders furnish standard sizes of receiver, from which selection may be made for any given case.

CHAPTER X

PERFORMANCE OF AIR COMPRESSORS

THE duty of air compressors may be designated in three ways:

First.—A useful standard of rating for ordinary purposes is the volumetric output, in terms of cubic feet of free air compressed per minute to a given pressure. The theoretical output or piston displacement (Table I) is found by multiplying the cylinder cross-section in square feet by the piston travel in feet per minute. The actual output is less than this on account of losses due to clearance, volume occupied by the piston rod, leakage, induction of warm air, and friction of inlet valves. In a well-designed compressor an allowance of 10 to 12 per cent will cover these losses, which must not be confused with the work required to overcome engine friction, and the added work due to heating of the air while being compressed (dealt with later).

Having found the volumetric capacity of the compressor, the volume V' of this air, at any given pressure P', is calculated by Boyle's formula (Chapter II): $V' = \frac{VP}{P'}$,

where V = initial volume of free air, cubic feet;
P = normal pressure of atmosphere (14.7 lb.);
P' = terminal absolute pressure = gage pressure + 14.7 lb.

For example, 100 cu.ft. of free air, compressed isothermally to 65 lb. gage, will occupy a volume:

$$V' = \frac{100 \times 14.7}{65 + 14.7} = 18.45 \text{ cu.ft.}$$

Conversely, the volume of free air corresponding to 18.45 cu.ft. of air at 65 lb. gage pressure is:

$$V = \frac{V'P'}{P} = \frac{18.45(65 + 14.7)}{14.7} = 100 \text{ cu.ft.}$$

By applying the 10 to 12 per cent allowance for losses stated above, sufficiently accurate results are obtained for most practical purposes.

TABLE I.—APPROXIMATE DISPLACEMENT, CUBIC FEET PER MINUTE

Double-acting Pistons in Single Cylinders at Various Piston Speeds: Volumes Occupied by Piston Rods Not Considered

Diam. of cyl., in.	Piston Speed, Feet per Minute													
	225	250	275	300	325	350	375	400	425	450	475	500	550	600
	Displacement, Cubic Feet per Minute													
5	30.8	34.2	37.5	41	44.5	48	51.4	54.4	58	61.6	65	68.4	75	82
6	44.4	49	54	59	64	69	74	78.4	84	88.8	93	98	108	118
7	60.5	67	74	80	87	94	100.6	106.8	114	121	127	134	148	160
8	79	87.5	96	105	114	123	132	140	149	158	166	175	192	210
9	100	110	122	133	144	155	166	176	188	200	210	220	244	266
10	123	136	150	164	178	192	206	218	233	246	260	272	300	328
11	149	165	182	198	215	232	250	264	282	298	315	330	364	396
12	177	196	216	236	256	276	296	312	335	354	374	392	432	472
13	208	230	255	276	300	325	345	368	392	416	440	460	510	552
14	240	266	295	320	348	375	400	424	455	480	510	532	590	640
15	276	305	337	368	400	432	460	488	521	552	580	610	674	736
16	315	350	385	420	455	490	525	556	595	630	660	700	790	840
17	355	395	435	472	512	550	509	628	672	710	750	790	870	944
18	400	440	485	530	575	620	660	704	755	800	840	880	970	1,060
19	445	490	540	590	640	690	740	784	840	890	930	980	1,080	1,180
20	490	545	600	656	710	765	820	872	930	980	1,040	1,090	1,200	1,312
22	595	660	725	790	860	930	990	1,060	1,120	1,190	1,260	1,320	1,450	1,580
24	710	790	865	940	1,020	1,100	1,180	1,256	1,340	1,420	1,500	1,580	1,730	1,880
26	830	920	1,020	1,100	1,200	1,300	1,390	1,476	1,570	1,660	1,750	1,840	2,040	2,200
28	900	1,070	1,180	1,280	1,390	1,500	1,600	1,780	1,820	1,920	2,030	2,140	2,360	2,560
30	1,110	1,230	1,350	1,480	1,600	1,720	1,840	1,960	2,100	2,220	2,340	2,460	2,700	2,960
32	1,260	1,400	1,540	1,670	1,820	1,960	2,100	2,240	2,380	2,520	2,660	2,800	3,080	3,340
34	1,420	1,570	1,740	1,890	2,050	2,220	2,360	2,520	2,690	2,840	3,000	3,140	3,480	3,780
36	1,600	1,770	1,950	2,120	2,300	2,500	2,650	2,840	3,000	3,200	3,360	3,540	3,900	4,240
38	1,780	1,960	2,170	2,360	2,560	2,750	2,950	3,120	3,350	3,560	3,750	3,920	4,340	4,720
40	1,970	2,180	2,400	2,620	2,850	3,060	3,280	3,480	3,720	3,940	4,150	4,360	4,800	5,240

As the volumetric output of a given size of cylinder depends on the density of the intake air, it will obviously be reduced when working at an altitude above sea-level (Chapter XII).

Second.—The size of the compressor may be designated in terms of the horsepower developed by the steam end, indicator cards being taken while running at normal working speed and while the usual volume of air is being compressed.

Third.—The horsepower represented by the quantity of compressed air delivered is determined from an indicator card taken from the air cylinder. In testing a compressor it is customary to take a series of cards, simultaneously from both ends of the steam and air cylinders.

If indicator cards are not available, the theoretical horsepower for single-stage adiabatic compression may be calculated by the formula (Eq. 16, Chapter II):

$$\text{Hp.} = \frac{144PVn}{33{,}000(n-1)}\left[\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} - 1\right]$$

in which P = normal atmospheric pressure, 14.7 lb. per sq.in.;

P' = final absolute pressure, lb. per sq.in.;

V = the volume of free air compressed per min., cu.ft.;

n = exponent of the compression curve.

For adiabatic compression, $n = 1.406$, and varies down to 1.18 or 1.2, according to the efficiency of the cooling arrangements, and whether single or stage compression. For the best single-stage compressors, $n = 1.3$ (approx.).

For isothermal compression (Eq. 4, Chapter II):

$$\text{Hp.} = \frac{144}{33{,}000} \times PV\left(\text{Nap. log}\,\frac{P'}{P}\right)^{*}$$

Table II shows the horsepowers required, under the conditions named therein, to compress one cubic foot of free air.

In columns 3 and 4 of Table II are given the theoretical horsepowers required for isothermal and adiabatic compression. The results of isothermal compression are wholly unattainable in practice, and are placed here only for comparison. The figures in column 4 are based on the assumptions that there is no radiation of heat from the air cylinder, and that the temperature of the air after delivery has become

* The Napierian or hyperbolic logarithm of a number is equal to the common logarithm multiplied by the constant 2.302585.

TABLE II.—SINGLE-STAGE COMPRESSORS

Gage pressure, lb.	Atmospheres absolute, or ratio of compression $\frac{P'}{P}$	Single-stage Compression, from Atmospheric Pressure at Sea-level. Initial Temp., 60° F. Horsepower Required to Compress 1 Cu.ft. of Free Air: Theoretical horsepower: Isothermal compression	Adiabatic compression	Actual horsepower (approx.): Allowance for losses above adiabatic compression, 10 per cent	Allowance for losses above adiabatic compression, 15 per cent
20	2.36	0.0551	0.0626	0.0689	0.0720
25	2.71	.0637	.0741	.0815	.0852
30	3.04	.0713	.0843	.0927	.0970
35	3.38	.0782	.0941	.1035	.1082
40	3.72	.0842	.1029	.1132	.1183
45	4.06	.0895	.1115	.1226	.1282
50	4.40	.0950	.1191	.1310	.1370
55	4.74	.0994	.1269	.1396	.1460
60	5.08	.1041	.1337	.1471	.1537
65	5.42	.1081	.1401	.1541	.1610
70	5.76	.1123	.1468	.1615	.1690
75	6.10	.1162	.1535	.1688	.1765
80	6.44	.1195	.1591	.1750	.1830
85	6.78	.1224	.1651	.1816	.1900
90	7.12	.1256	.1703	.1873	.1955
95	7.46	.1287	.1760	.1936	.2024
100	7.80	.1315	.1807	.1988	.2080
110	8.48	.1366	.1894	.2083	.2180
125	9.50	.1442	.2025	.2227	.2328

normal, its volume being therefore reduced to that which is practically available for use. These figures include no losses except those due to the heating of the air while being compressed. But the full amount of loss represented by adiabatic compression can never be suffered in the operation of compressors, however imperfect their design. The actual compression line is always lower than the adiabatic line, because of the radiation of heat through the cylinder walls. Even in single-stage compressors, properly water-jacketed and run at a reasonable piston speed, the compression line may fall considerably below the adiabatic. Whatever diminution of loss is effected by cooling the air in the cylinder may therefore be credited against the other unavoidable losses, partially offsetting them: frictional or mechanical loss in the compressor, friction of inlet valves, heating of the intake air by contact with the hot metal surfaces, and piston clearance.

In the absence of indicator cards, estimates based on practice may be made of the compressor horsepower. In columns 5 and 6 of Table II are shown estimated horsepowers required to compress 1 cu.ft. of free air, under the conditions stated at top of the columns. Thus, in column 5, 10 per cent is assumed as a fair estimate, in case of large, well-designed and operated single-stage compressors in good running order, of the additional power required, over and above that for adiabatic compression. This 10 per cent is taken as the algebraic sum of: the loss in purely adiabatic compression, minus the effect of ordinary water-jacket cooling, plus the four losses mentioned at the end of the preceding paragraph. In column 6, the power consumed in adiabatic compression is increased by 15 per cent, which represents relatively poorer work. (See also Table III.)

It must be remembered that, if the compressor is small, or in poor condition, or is run at too high a speed, the required horsepower is greater. In such cases, the added percentages in column 6 of Table II and columns 5 and 7 of Table III may be increased to 18 or 20 per cent.

The figures in columns 3 and 4 or 5 and 6 of Table II (which are for *free* air), if multiplied by the corresponding ratios of compression (column 2), will give the respective theoretical and actual power costs of furnishing 1 cu.ft. of *compressed* air, at the gage presssure stated. These figures have been checked against the results obtained from a number of representative compressors in operation, and will be found to approximate closely the actual power costs under normal, average conditions.

Work of Stage Compressors (see equations 19 and 20, Chapter II). The theoretical horsepower for two-stage compression is:

$$\text{Hp.} = \frac{2 \times 144}{33{,}000} \times \frac{PVn}{n-1}\left[\left(\frac{P'}{P}\right)^{\frac{n-1}{2n}} - 1\right]$$

For three-stage compression:

$$\text{Hp.} = \frac{3 \times 144}{33{,}000} \times \frac{PVn}{n-1}\left[\left(\frac{P'}{P}\right)^{\frac{1-n}{3n}} - 1\right]$$

Reducing the constants, and for a volume of 1 cu.ft. free air:

$$\text{Two-stage,}\quad \text{Hp.} = 0.444\left[\left(\frac{P'}{P}\right)^{0.141} - 1\right]$$

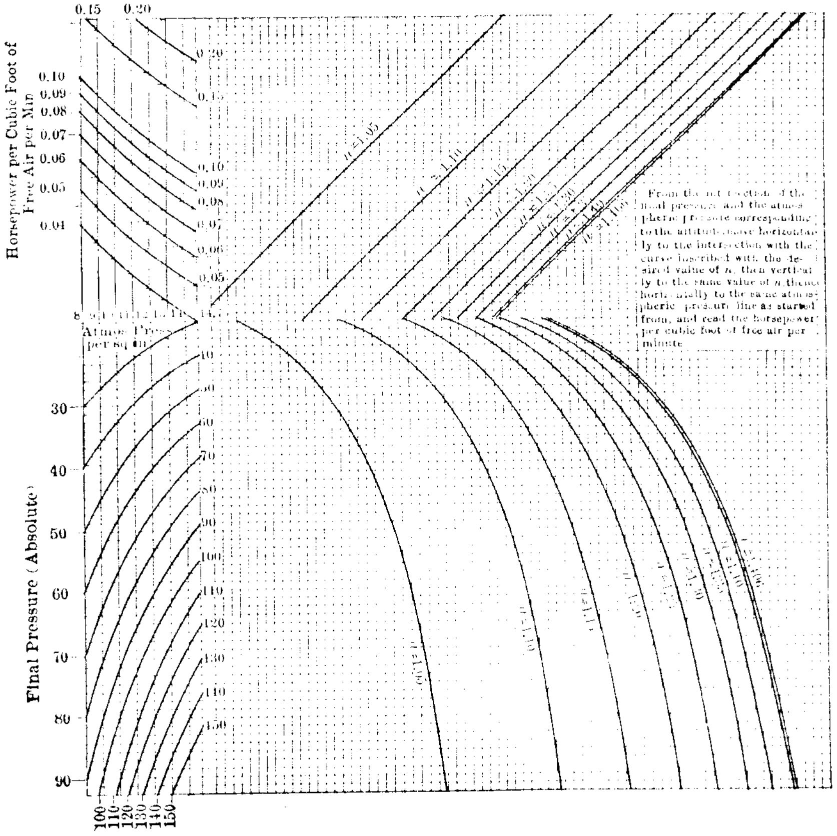
$$\text{Three-stage, Hp.} = 0.6663\left[\left(\frac{P'}{P}\right)^{0.0962} - 1\right]$$

TABLE III.—TWO- AND THREE-STAGE COMPRESSORS

Gage pressure, lb.	Ratio of compression $\frac{P'}{P}$	Horsepower per Cubic Foot of Free Air at Sea-level				
		Isothermal compression	Two-stage compression		Three-stage compression	
			Adiabatic compression	Actual Hp. on basis of adiabatic compression +15 per cent	Adiabatic compression	Actual Hp. on basis of adiabatic compression +12 per cent
70	5.76	0.1123	0.129	0.148		
80	6.4	.1195	.138	.159		
90	7.12	.1256	.147	.169		
100	7.80	.1315	.154	.177	0.145	0.162
120	9.16	.1420	.169	.194	.158	.177
140	10.50	.1508	.181	.208	.169	.189
160	11.88	.1583	.192	.221	.179	.200
180	13.24	.1654	.202	.232	.188	.210
200	14.60	.1720	.212	.241	.196	.219
250	18.00	.1853	.231	.266	.213	.238
300	21.40	.1963	.249	.285	.228	.255
350	24.80	.2058	.264	.303	.241	.270
400	28.20	.2140	.277	.318	.252	.282
450	31.62	.2215	.289	.332	.262	.293
500	35.01	.2280			.271	.303
550	38.41	.2339			.280	.313
600	41.80	.2393			.288	.322
650	45.21	.2443			.295	.330
700	48.62	.2490			.301	.337
800	55.42	.2574			.314	.352

Tables II and III may be used for any altitude above sea-level, by considering the ratio of compression in column 2. Thus, to find the horsepower for two-stage compression per cubic foot of free air at 10,000 ft. altitude and 80 lb. gage: Ratio of compression $= \frac{80 + 10.70}{10.07} = 8.94$ (the factor 10.07 being the barometric pressure at 10,000 ft., taken from Table I, Chapter XII). From Table III, column 2, 8.94 compressions (by interpolation) correspond to a gage pressure of about 118 lb. at sea-level, the horsepower for which (column 5) equals 0.191 per cubic foot of free air. At sea-level, the horsepower would be 0.159.

Graphic Determination of the horsepower required to compress 1 cu.ft. of free air to a given pressure (C. W. Crispell, *Trans.* Am. Inst. Min. Engs., Vol. 58, p. 93). For this the nomograms in Figs. 1 and 2 are more convenient than the formulas. The values of P (atmos. pressure) in pounds per square inch, for different altitudes, are given in Table I, column 3 (Chapter XII). Directions for using the charts are printed on them. Note that the scale for the horsepower per cubic foot of air, in Figs. 1 and 2, begins at a different ordinate for each value of P. By a series of examples in the paper quoted, it is shown that the nomograms usually give results from 0.7 to 1.6 per cent higher than those from the formulas. The diagram in Fig. 3 will also be found useful.



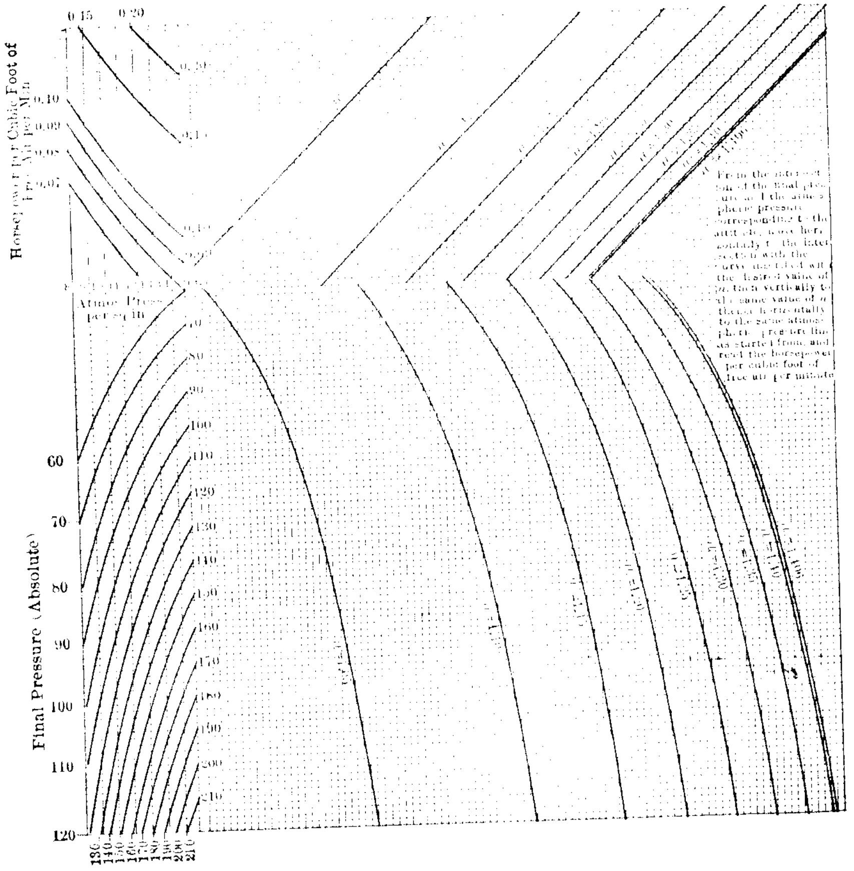
From the intersection of the final pressure and the atmospheric pressure corresponding to the altitude, move horizontally to the intersection with the curve inscribed with the desired value of n, then vertically to the same value of n, thence horizontally to the same atmospheric pressure line as started from, and read the horsepower per cubic foot of free air per minute.

FIG. 1.—Single-stage Compression.

Efficiency of Compressors may be designated in four ways:

(1) *Volumetric efficiency*, ordinarily say 65 to 80 per cent, is the ratio of the volume of compressed air delivered to the volume of cylinder dis-

placement. It is useful in determining the output of air available for operating drills and other machines using the air, but does not necessarily connote economical performance of the compressor. A compressor having high volumetric efficiency may have low total efficiency, and conversely. For example, a relatively low volumetric efficiency may merely indicate that the air ports and valves are of large area, with consequent small pressure losses and high total efficiency. Comparisons should therefore be based upon the horsepower per cubic foot of air output.

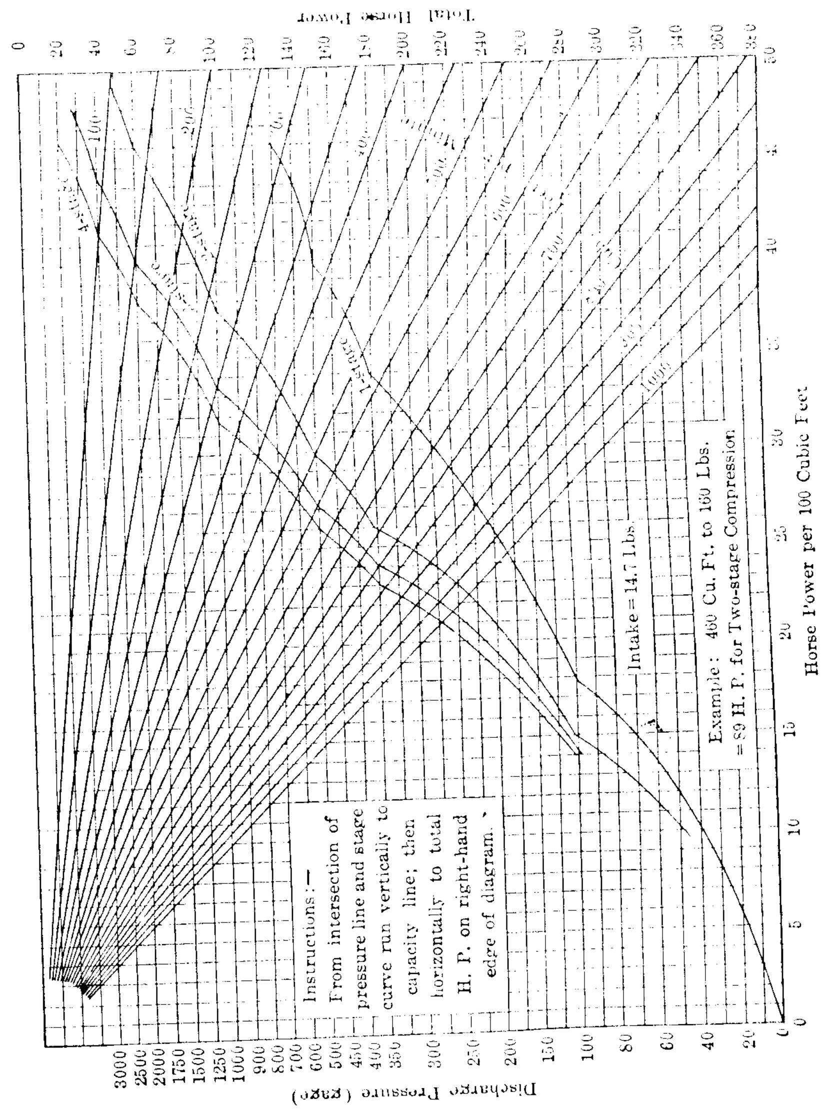


From the intersection of the final pressure and the atmospheric pressure corresponding to the altitude, trace horizontally to the intersection with the curve marked with the desired value of n, then vertically to the same value of n, thence horizontally to the same atmospheric pressure line as started from, and read the horsepower per cubic foot of free air per minute.

FIG. 2.—Two-stage Compression.

(2) *Compression efficiency* is the ratio of the theoretical power required to compress the volume of air delivered to the actual indicated horsepower developed in the air cylinder. According to the mode of compression, it is designated as Isothermal (ordinarily say

72 to 75 per cent) or Adiabatic Compression Efficiency (say 80 to 86 per cent). It may be found also by using the mean effective pressure in the air cylinder (Table IV). For a stage compressor the mean effective pressure is the total referred to the low-pressure cylinder.



Instructions:—
From intersection of pressure line and stage curve run vertically to capacity line; then horizontally to total H. P. on right-hand edge of diagram.

Example: 460 Cu. Ft. to 160 Lbs. = 89 H. P. for Two-stage Compression

FIG. 3.—Horsepower Chart Adiabatic Compression, Perfect Intercooling. S. H. Redfield.

(3) *Mechanical efficiency* (ordinarily 90 to 95 per cent) is the ratio of the indicated horsepower represented by the air delivered to the indicated horsepower of the steam (for a steam-driven compressor),

or to the brake horsepower (for a power-driven compressor). The difference between the two is the power consumed in overcoming the frictional resistance of the compressor.

(4) *Total or overall efficiency* (roughly 75 per cent) is the adiabatic compression efficiency (see (2) above), multiplied by the mechanical efficiency. It is the most important from the standpoint of the user of the compressor, because it is the ratio of the power represented by the volume of compressed air delivered to the power consumed in producing it. For a steam-driven compressor, the total efficiency is as stated above; for a power-driven compressor, it is the compression efficiency × mechanical efficiency × overall efficiency of the prime mover (including belting). In using this efficiency it should be understood that, for the same conditions, it is higher when referred to the adiabatic base than when referred to the isothermal base.

The following formulas (S. B. Redfield in "Compressed Air Data") exhibit in condensed form the relations between the factors entering into the general expressions for mechanical, volumetric and overall efficiencies. The terms Isothermal Factor and Adiabatic Factor, as employed in expressing Overall Efficiency, are the theoretical horsepowers required to compress isothermally or adiabatically 100 cu.ft. of free air per minute to the designated pressure. For computations involving differences in barometric pressure at altitudes it is now customary to use the lower base, represented by the Isothermal Factors (Chapter XII).

$$\frac{\text{Indicated Hp. of Air Cyls.}}{\text{Indicated Hp. of Steam Cyls.}} = \text{Mechanical Efficiency.}$$

$$\frac{\text{Length of Intake Line, Low-press. Cyl.}}{\text{Total Length of Indicator Diagram}} = \text{Indicated Volumetric Efficiency.}$$

$$\frac{\text{Cu.Ft. Free Air Delivered}}{\text{Cu.Ft. Piston Displacement}} = \text{Actual Volumetric Efficiency.}$$

$$\frac{\text{Actual Volumetric Efficiency}}{\text{Indicated Volumetric Efficiency}} = \text{Slippage Efficiency.}$$

$$\frac{\text{Isothermal Factor (see above)}}{\text{I.Hp. of Steam Cylinders per 100 Cu.Ft. Free Air Del'd per Min.}}$$

$$= \text{Overall Isothermal Compression Efficiency.}$$

$$\frac{\text{Aver. Actual Work Done}}{\text{Max. Capacity for Work}} = \text{Load Factor of Compressor}$$

Additional formulas for electric motor-driven compressor:

$$\frac{\text{Brake Hp}}{\text{Electrical Hp.}} = \text{Motor Efficiency.}$$

$$\frac{\text{I.Hp. of Air Cyls.}}{\text{Brake Hp.}} = \text{Mechanical Efficiency.}$$

$$\frac{\text{Input of Elec. Hp.} \times 100}{\text{Cu.Ft. Air Del'd per Min.}}$$

$$= \text{Elec. Hp. per 100 Cu.Ft. Free Air Del'd per Min.}$$

$$\frac{\text{Isothermal Factor}}{\text{Elec. Hp. per 100 Cu.Ft. Free Air Del'd per Min.}}$$

$$= \text{Overall Isothermal Compression Efficiency}$$

Table IV is useful for calculations based on volumes and mean cylinder pressures. The mean pressures per stroke (columns 5 and 6) are obtained from the formulas for isothermal and adiabatic single-stage compression, by making $V = 1$, thus:

$$\text{Mean pressure per stroke (isothermal)} = P \times \text{Nap.}\log\frac{P'}{P}$$

$$\text{Mean pressure per stroke (adiabatic)} = 3.463P\left[\left(\frac{P'}{P}\right)^{0.29} - 1\right]^*$$

The work done during one stroke is equal to the mean pressure multiplied by the volume in cubic feet traversed by the piston.

When air is compressed adiabatically, the relation between its temperature T, at the beginning of compression, and the terminal temperature T', is shown by:

$$\frac{T'}{T} = \left(\frac{V}{V'}\right)^{n-1}, \quad \text{whence} \quad T' = T\left(\frac{V}{V'}\right)^{n-1}$$

The final temperature may also be found from the formula:

$$T' = T\left(\frac{P'}{P}\right)^{\frac{n-1}{n}}$$

T and T' being absolute temperatures and P, P' absolute pressures.

The compression curve of an air-indicator card may be constructed

---

* The constant $3.463 = \frac{n}{n-1} = \frac{1.406}{.406}$; exponent $0.29 = \frac{n-1}{n} = \frac{1.406-1}{1.406}$.

Table IV*

Gage pressures	Atmospheres	Volume with air at constant temperature (isothermal)	Volume with air not cooled (adiabatic)	Mean pressure per stroke; air at constant temperature, lb.	Mean pressure per stroke; air not cooled, lb.	Temperature of air; not cooled. Deg. F.
0	1	1	1	0	0	60°
1	1.068	.9363	.9500	.96	.975	71
2	1.136	.8803	.9100	1.87	1.91	80.4
3	1.204	.8305	.8760	2.72	2.80	88.9
4	1.272	.7861	.8400	3.53	3.67	98
5	1.340	.7462	.8100	4.30	4.50	106
10	1.680	.5952	.6900	7.62	8.27	145
15	2.020	.4950	.6060	10.33	11.51	178
20	2.360	.4237	.5430	12.62	14.40	207
25	2.700	.3703	.4940	14.59	17.01	234
30	3.040	.3289	.4538	16.34	19.40	252
35	3.381	.2957	.4200	17.92	21.60	281
40	3.721	.2687	.3930	19.32	23.66	302
45	4.061	.2462	.3700	20.57	25.59	321
50	4.401	.2272	.3500	21.69	27.39	339
55	4.741	.2109	.3310	22.76	29.11	357
60	5.081	.1968	.3144	23.78	30.75	375
65	5.423	.1844	.3010	24.75	32.32	389
70	5.762	.1735	.2880	25.67	33.83	405
75	6.102	.1639	.2760	26.55	35.27	420
80	6.442	.1552	.2670	27.38	36.64	432
85	6.782	.1474	.2566	28.16	37.94	447
90	7.122	.1404	.2480	28.89	39.18	459
95	7.462	.1340	.2400	29.57	40.40	472
100	7.802	.1281	.2320	30.21	41.60	485
105	8.142	.1228	.2254	30.81	42.78	496
110	8.483	.1178	.2189	31.39	43.91	507
115	8.823	.1133	.2129	31.98	44.98	518
120	9.163	.1091	.2073	32.54	46.04	529
125	9.503	.1052	.2020	33.07	47.06	540
130	9.843	.1015	.1969	33.57	48.10	550
135	10.183	.0981	.1922	34.05	49.10	560
140	10.523	.0950	.1878	34.57	50.02	570
145	10.864	.0921	.1837	35.09	51.00	580
150	11.204	.0892	.1796	35.48	51.89	589
160	11.880	.0841	.1722	36.29	53.65	607
170	12.560	.0796	.1657	37.20	55.39	624
180	13.240	.0755	.1595	37.96	57.01	640
190	13.920	.0718	.1540	38.68	58.57	657
200	14.600	.0685	.1490	39.42	60.14	672

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* From a table by Frank Richards.

as follows, PV being the pressure and volume at one point of the curve and $P'V'$ the pressure and volume corresponding to any other point. Designating the index number of the curve by x:

$$\frac{P}{P'} = \left(\frac{V'}{V}\right)^x.$$

From this, $\log\left(\frac{P}{P'}\right) = x \log\left(\frac{V'}{V}\right)$; whence, $x = \dfrac{\log\left(\frac{P}{P'}\right)}{\log\left(\frac{V'}{V}\right)}$

The several lines of an air card have significations entirely different from those of a steam card. In Fig. 4 (an ideal card), AB is the admission line, BC the compression line, CD the delivery or discharge line, and DA the re-expansion line. DA represents the effect of the re-expansion of the clearance air, on beginning a stroke. Comparing the lines of the air and steam cards, they are found to be reversed:

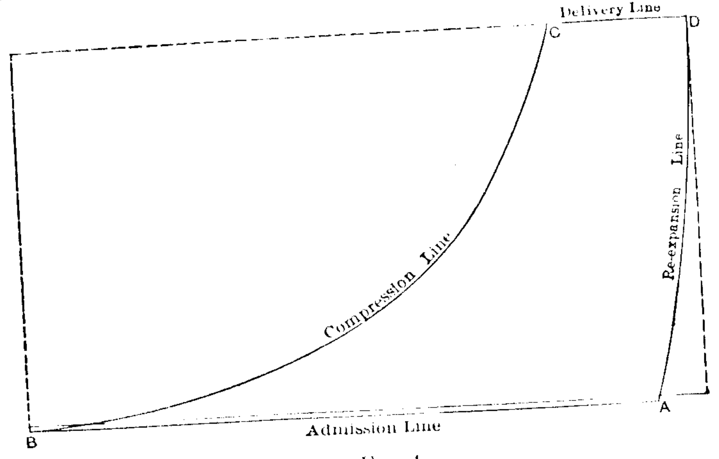

Fig. 4.

Air Card	Steam Card
Admission line	Back-pressure or exhaust line
Compression line	Expansion line
Delivery line	Admission line
Re-expansion line	Compression line

The elements of an air card, together with the work done, as represented by the several lines and areas, are further elucidated by Fig. 5, the compression being adiabatic.

Let AD = normal atmospheric line at sea-level;
$AG = P$ = corresponding atmospheric pressure, acting behind the piston at the beginning of the stroke (neglecting valve resistances and effect of clearance of previous stroke);
$GE = AD$ = length of stroke of piston;
AB = adiabatic compression curve;
BC = delivery line.

At the point B the useful work of compression ceases; during the remainder of the stroke the volume of compressed air V', at the absolute pressure P', is being forced out of the cylinder through the delivery valves.

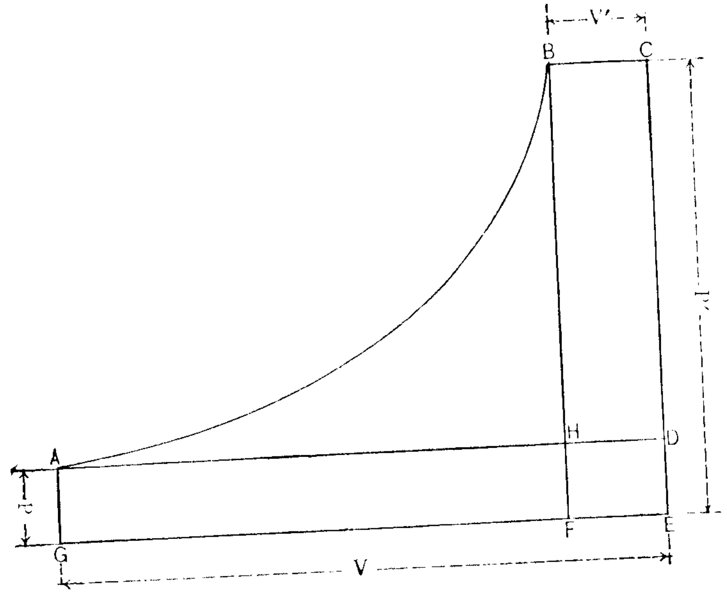

FIG. 5.

The area $ABFG$ = the absolute work of compression.
The area $BCEF$ = the absolute work of delivery.
The sum of these areas represents the total absolute work (that is, on the basis of absolute pressure) done during compression and delivery.
Area $ADEG$ = work done during the entire stroke by atmospheric pressure behind the piston.
Area ABH = net work of compression.
Area $BCDH$ = net work of delivery.
Area $ABCDA$ = total net work for entire stroke.

From this analysis another method may be derived for calculating the theoretical horsepower required for compressing air. It will be found useful, when a table of temperatures of compression is available.

Let w = weight of 1 cu.ft. of free air = 0.0765 lb.;
C_p = specific heat of air at constant pressure = 0.2375;
C_v = specific heat of air at constant volume = 0.1689.

$$\text{Whence,} \quad \frac{C_p}{C_v} = n = 1.406, \quad \text{and} \quad \frac{n}{n-1} = 3.463.$$

J = Joule's heat unit, taken as 778 ft.-lb.

The work represented by the area ABH =

$$J \times w \times C_v(T' - T) - P(V - V').$$

Also, the work done during delivery = $BCDH = V'(P' - P)$. Hence, the total net work for one stroke of the piston

$$= \text{area } ABCDA = J \times w \times C_v(T' - T) - (PV - P'V').$$

If C_p is substituted for C_v, then $PV = P'V'$, according to the general equation for air compression, and the total work

$$W = J \times w \times C_p(T' - T).$$

Substituting for J, w, and C_p, their constant numerical values:

$$W = 14.13(T' - T),$$

or, to compress 1 cu.ft. of air per min., at 60° F., and at sea-level

$$\text{Hp.} = 0.225\left[\frac{T'}{T} - 1\right].$$

By referring to the last column of Table IV and remembering that T and T' are absolute temperatures (thermometric temperatures plus 459° F.), the horsepower required for compressing 1 cu.ft. of free air adiabatically to any gage pressure may readily be calculated.

Other expressions for the mean effective pressures may also be deduced from what precedes. For the entire stroke,

$$\text{M.E.P.} = P\frac{n}{n-1}\left(\frac{T'}{T} - 1\right) = 3.46P\left(\frac{T'}{T} - 1\right) = 3.46P\left[\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} - 1\right]$$

During delivery

$$\text{M.E.P.} = \frac{V'}{V}(P' - P).$$

The M.E.P. for compression only is found by taking the difference between the pressures calculated by the last two formulas.

The results from the above expressions for work and mean effective pressure are theoretical. To find the actual horsepower required, allowances must be made for the losses experienced in the operation of the compressor, as already set forth.

Compressor Tests.—In selling a compressor, the maker usually guarantees a specified volumetric efficiency. As this is important from the user's standpoint, it would be desirable if all makers employed the same method of testing, so that accurate comparisons could be drawn. The volume of air actually available is best determined by the low-pressure conoidal orifice, or by the shield type of the F.L.M. meter, largely used on the Witwatersrand, South Africa. For details of both methods, with the corrections for temperature, barometric pressure and orifice coefficients, see Chapter XXV.

A complete mechanical test is a much more elaborate operation, especially if the compressor is steam-driven. The variable conditions of service must be ascertained, with the deductions from observed data, to determine the power input and the output in terms of air delivered at specified piston speeds.

Details of a complete test of a compound, two-stage Nordberg compressor, of the Tennessee Copper Co., will be found useful.* Items 28, 29, and 32 to 35, were necessary, because the boilers supplied steam for a hoisting engine and an independent condenser, as well as for the compressor. Though the hoist was not running, steam passed continuously to the jackets of its cylinders. Similar conditions are often met in other tests. The boiler feed-water was taken during the run from two barrels set on scales. The water of condensation from steam jackets and reheater was drawn off continuously and also weighed. Calorimeter tests were made with a Peabody throttling calorimeter. Eight sets of indicator cards were taken during the 8-hour test, at hourly intervals.

ITEMS OF COMPRESSOR TEST (Altitude, 1,800 ft.)

1. Date of test, February 16, 1902	
2. Duration of test, hours	8
3. Diameter of high-pressure steam cylinder (steam jacketed), inches	14
4. Diameter of low-pressure steam cylinder (steam jacketed), inches	28
5. Diameter of low-pressure air cylinder, inches	$24\frac{1}{2}$
6. Diameter of high-pressure air cylinder, inches	$15\frac{3}{8}$
7. Stroke of all pistons, inches	42
8. Diameter of piston rods, inches	$2\frac{11}{16}$
9. Revolutions of engine, average per minute	90
10. Piston speed per minute, feet	630
11. Steam-gage pressure, average, pounds	145.9
12. Temperature of steam in steam pipe, average, degrees F.	364
13. Steam pressure in reheating receiver, average, pounds	8
14. Vacuum in condenser, average, inches	25.66
15. Air pressure in intercooler, average, pounds	22.63
16. Air pressure in receiver, average, pounds	79.3
17. Temperature of air at intake, average, degrees F	65.0
18. Temperature of air leaving low-pressure cylinder, average, degrees F.	211.5
19. Temperature of air leaving intercooler, average, degrees F.	78.5
20. Temperature of air leaving high-pressure cylinder, average, degrees F.	240.0
21. Indicated horsepower in high-pressure steam cylinder, average	140.12
22. Indicated horsepower in low-pressure steam cylinder, average	153.03
23. Indicated horsepower in both steam cylinders, average	293.15
24. Indicated horsepower in low-pressure air cylinder, average	143.79
25. Indicated horsepower in high-pressure air cylinder, average	135.02
26. Indicated horsepower in both air cylinders, average	278.81
27. Feed-water weighed to boilers, pounds	43,343
28. Re-heater and jacket water from compressor, weighed, pounds	4,081
29. Average temperature of re-heater and jacket water, degrees F.	356.7
30. Total heat in 1 pound of steam at 356.7 degrees F., heat units	1,190.7
31. Total heat in 1 pound of water at 356.7 degrees F., heat units	328.9
32. Equivalent credit for re-heater and jacket water, pounds	1,127.00
33. Water weighed from condensation in hoisting-engine jacket, pounds	1,781.00
34. Steam used to run condenser, pounds	4,300.00
35. Total credits to feed-water, pounds	7,228.00
36. Total feed-water charged to engine, pounds	36,115.00
37. Moisture in steam shown by Peabody calorimeter, per cent	1.30
38. Credit for moisture in steam, pounds	473.00
39. Total steam charged to engine, pounds	35,642.00
40. Dry steam per hour charged to engine, pounds	4,455.00
41. Steam consumption per indicated horsepower per hour, pounds	15.19
42. Guaranteed steam consumption per indicated horsepower per hour, at 92 revolutions per minute, pounds	14.00
43. Excess of steam consumption per indicated horsepower per hour over guarantee, pounds	1.19
44. Theoretical delivery of free air per minute at 90 revolutions, cubic feet	2,037.8
45. Slip of air (percentage)	3.0
46. Actual slip of air per minute, cubic feet	61.1
47. Actual delivery of free air per minute, average, cubic feet	1,976.7
48. Theoretical horsepower required to compress and deliver actual delivery of air at receiver pressure by adiabatic compression	306.53
49. Theoretical horsepower required to compress and deliver actual delivery of air at receiver pressure by isothermal compression	229.00
50. Actual horsepower shown by air indicator cards	278.81
51. Actual horsepower shown by steam indicator cards	293.15
52. Actual horsepower consumed by friction of engine	14.34
53. Efficiency ratio between steam and air cylinders, per cent	95.1
54. Efficiency ratio between steam and air cylinders guaranteed by builder, per cent	87.0
55. Efficiency of steam, or ratio of steam indicated horsepower to theoretical air indicated horsepower, isothermal compression, per cent	78.1

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* From an article by J. Parke Channing, *Mines and Minerals*, May, 1905, p. 475.

One of the combined indicator cards, from which the averages in items 21-26 were calculated, is shown in Fig. 6.

In further illustration of the performance of air compressors, see the cards in Figs. 7, 8 and 9.

It would tend to greater economy in the production of compressed air if superintendents and master mechanics gave more attention to the actual results from the compressors in their charge, and studied the unfavorable conditions under which they frequently operate.

Few records of the effective horsepower of compressors have been published. To express the efficiency, it is customary to divide the horsepower of the air cylinder by the horsepower of the steam cylinder, as determined by indicator cards. The maker's rating is often based on mechanical efficiency, considering no losses except those of friction. Such a criterion does not properly measure the relative commercial values of compressors, nor does it indicate the effective horsepower developed under normal conditions. See "Efficiency of Compressors," p. 193.

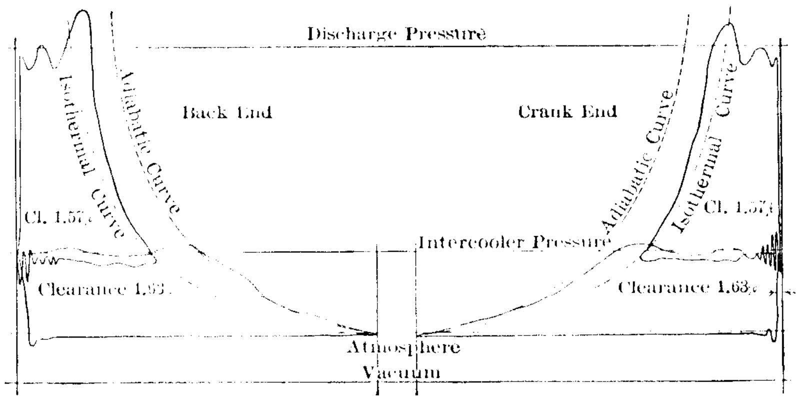

FIG. 6.—Combined Cards. Two-stage Nordberg Compressor.

Field Tests were made in 1909 by Richard L. Webb, consulting engineer, on a number of compressors in a well-known Canadian mining district. In conducting these tests, access was had to plants which had been in operation for a year or more under normal working conditions. As most of the compressors were in care of competent mechanics and in good order, the results present a fair average of current practice at mines.

Each test was conducted as follows:

First, a boiler test was run for at least two weeks, the coal being weighed, the feed water measured, and the total revolutions of the compressor recorded by a revolution counter. From these data, the

cost per boiler horsepower and the average compressor speed were determined. Second, the compressor was operated at different speeds over its entire range. By a meter in the steam pipe near the throttle, the total steam consumed, in pounds per hour, was measured. Indicator cards were taken of all cylinders, with temperatures at the air inlet, intercooler, and discharge. To measure the air delivered, a meter was placed in the discharge pipe from the receiver. From a series of simultaneous readings on all instruments, taken at each speed, were calculated the total horsepower of the steam and air cylinders, the steam consumption, and the total piston displacement per minute.

The air and steam meters, of the Dodge type, as modified by the General Electric Co., were operated by their own expert. The indica-

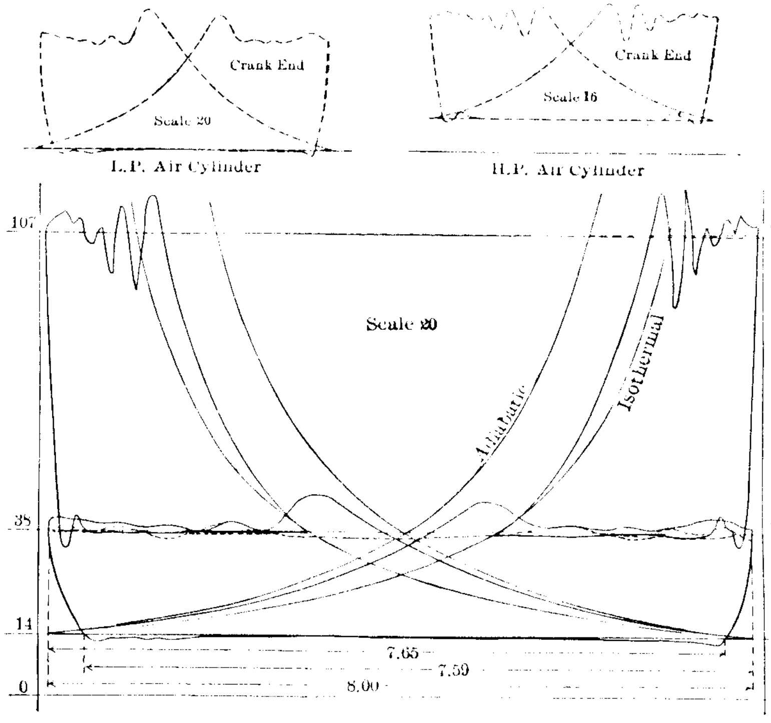

FIG. 7.—Combined Cards, Ingersoll-Rand, Two-stage, Direct-connected, Electrically-driven Compressor. Air Cylinders 23" and 14" × 20".

Rev. per min.	187.	I.Hp. of high-pressure cylinder	120.8
Piston speed, ft. per min.	623.3	Total I.Hp.	252.8
Discharge air pressure, lb.	93.	Free air delivered per min. cu.ft. (from card)	1,706.
Intercooler pressure, lb.	24.	Effic. compared with adiabatic	97.2%
Volumetric efficiency (from card)	95.3%	Effic. compared with isothermal	84%
I.Hp. of low-pressure cylinder	132.		

tors were the Roberts-Thompson and the American-Thompson, which are generally accepted as standard.

The friction loss was found to be a small item in the total. The other losses, often overlooked or disregarded, seriously reduced the efficiency. The mechanical efficiency only is too often considered. Thus, if the horsepower of the air cylinder is 100, and that of the steam cylinder 110, the compressor efficiency is rated at 91 per cent. This rating disregards losses due to adiabatic compression, heating of the cylinder and friction of the air passing through the valves. The tests showed the power loss due to engine friction to be usually not less than 10 per cent and often larger. Losses from the other causes mentioned ranged from 30 per cent up. Following are details of three representative tests.

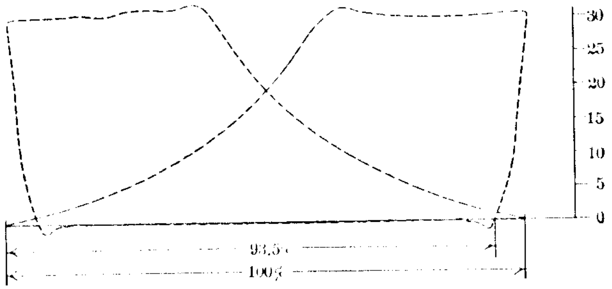

Fig. 8.—Card from 30¼ in. × 24 in. L. P. Air Cylinder of Style "O," Ingersoll-Rand Compressor. (Steam pressure, 115 lb.; air pressure, 28 lb.; r.p.m., 100; spring, 20.)

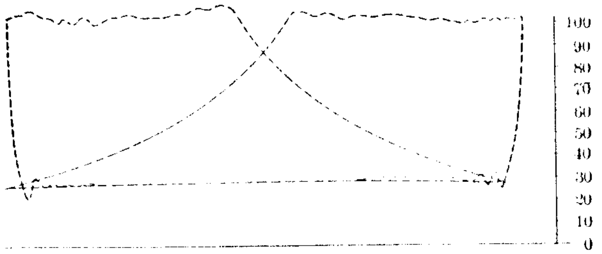

Fig. 9.—Card from 18¼ in. × 24 in. H.P. Air Cylinder, Style "O," Ingersoll-Rand Compressor. (Steam pressure, 115 lb.; air pressure, 100 lb.; r.p.m., 100; spring, 60.)

Plant One. Three 125-hp. boilers (one being held in reserve) supplied steam for a cross-compound, two-stage compressor; steam cylinders were of 16- and 28-in. diameter by 24-in. stroke; air cylinders, 28- and 18-in. by 24-in. stroke. Results from a two weeks' run were:

Total coal burned, pounds	264,300
Total feed-water, cubic feet	37,459
Total feed-water, pounds	2,335,568
Average temperature of feed water, degrees F.	131
Average evaporation per pound coal consumed, pound	8.72
Average revolutions per minute	63.1
Indicated horsepower of steam end, corresponding to 63.1 revolutions per minute	161
Corresponding indicated horsepower of air end	123
Average steam pressure, pounds	115
Average vacuum, pounds	10.5
Average air pressure, pounds	96
Average temperature of outside air, degrees F.	24
Average air piston displacement at 70° F., cubic feet	1,172
Average metered output corrected to 70° F., cubic feet	758

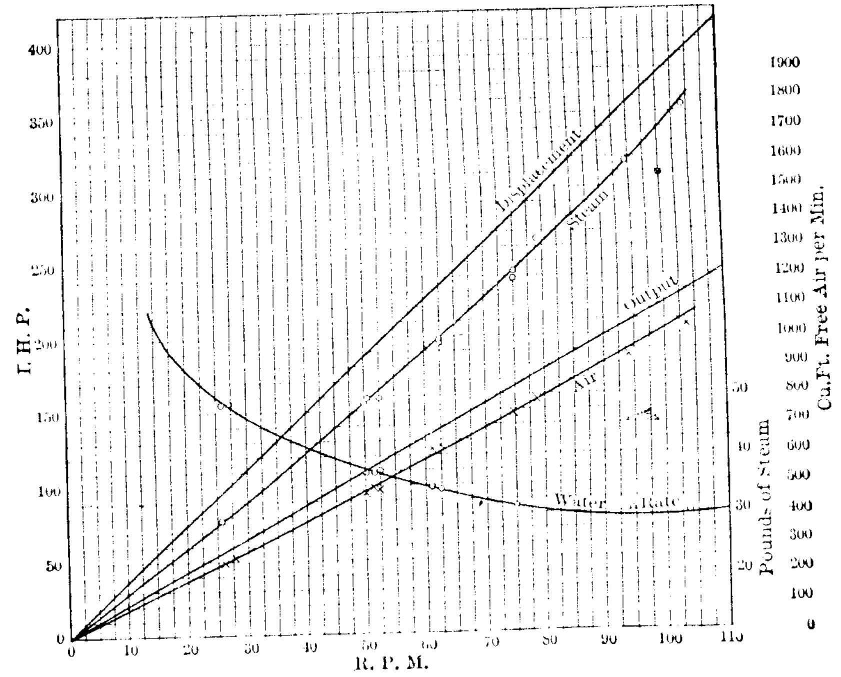

Fig. 10.—Compressor Plant No. 1.

The average evaporation, of 8.72 lb. of water from 131° F. to an average steam pressure of 115 lb., is equivalent to 9.83 lb. of water evaporated from and at 212° F. per pound of coal consumed. At the average speed of 63.1 r.p.m., the metered output was 758 cu.ft. of free air per minute, the piston displacement being 1,172 cu.ft. per minute.

Table V and Fig. 10 show the average operating results. In the figure, the curve at 63 revolutions are followed, at which the indicated

TABLE V.—TEST ON PLANT NUMBER ONE

Compound Condensing Steam Cylinders, 16 in. and 28 in., with Meyer Valve Gear; Air Cylinders, 28 in. and 18 in., Stroke 24 in.

Rev. per min.	Steam pressure, lb.	Air pressure, lb.	High-pressure Side				Low-pressure Side				Total steam, I.Hp.	Total air, I.Hp.	Friction loss	Friction loss, per cent	Air-line temperature	Cu.ft. displacement, 70° F.	Metered air output, cu.ft. per min., 70° F.	Calculated output, cu.ft.
			Steam		Air		Steam		Air									
			M.E.P.	I.Hp.	M.E.P.	I.Hp.	M.E.P.	I.Hp.	M.E.P.	I.Hp.								
25.8	125	75	55.0	34	33.95	20.68	10.3	19.75	14.31	21.68	53.75	42.36	11.39	21.2	220	480	340	295
32.7	126	87	63.2	49.5	39.65	30.65	10.7	26.0	14.69	28.2	75.5	58.85	16.65	22.05	245	610	415	375
43.0	126	95	63.5	65.45	42.44	43.10	11.65	37.2	14.16	35.73	102.7	78.8	23.9	23.2	250	800	519	492
55.5	129	100	64.6	86.0	46.15	60.4	13.61	56.2	15.32	49.9	142.2	110.3	31.9	22.4	258	1,030	605	635
64.0	128	100	65.65	100.7	48.25	72.8	15.1	71.8	15.85	59.5	172.5	132.3	40.2	23.3	268	1,190	650	730
73.0	126	94	61.2	107.1	42.88	73.8	14.78	80.2	15.64	67.0	187.3	140.8	46.5	24.8	262	1,355	690	835
88.0	125	97	61.62	129.9	48.4	100.5	17.12	112.0	16.56	86.7	241.9	186.2	55.7	23.0	286	1,655	740	898
101.08	127	95	63.85	155.9	47.75	114.7	16.5	124.9	15.8	94.5	280.8	209.2	71.6	25.5	280	1,890	760	1,165

NOTE.—The above data are from a table of fourteen readings taken at different speeds.

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horsepower of the steam cylinder was 160.8, and that of the air cylinder, 123, showing a mechanical efficiency of 76.5 per cent. The theoretical horsepower required to compress isothermally 1 cu.ft. of free air per minute to 96 lb. is 0.129; the useful work done by the compressor being 758 × 0.129 = 97.8, and the net total efficiency of the

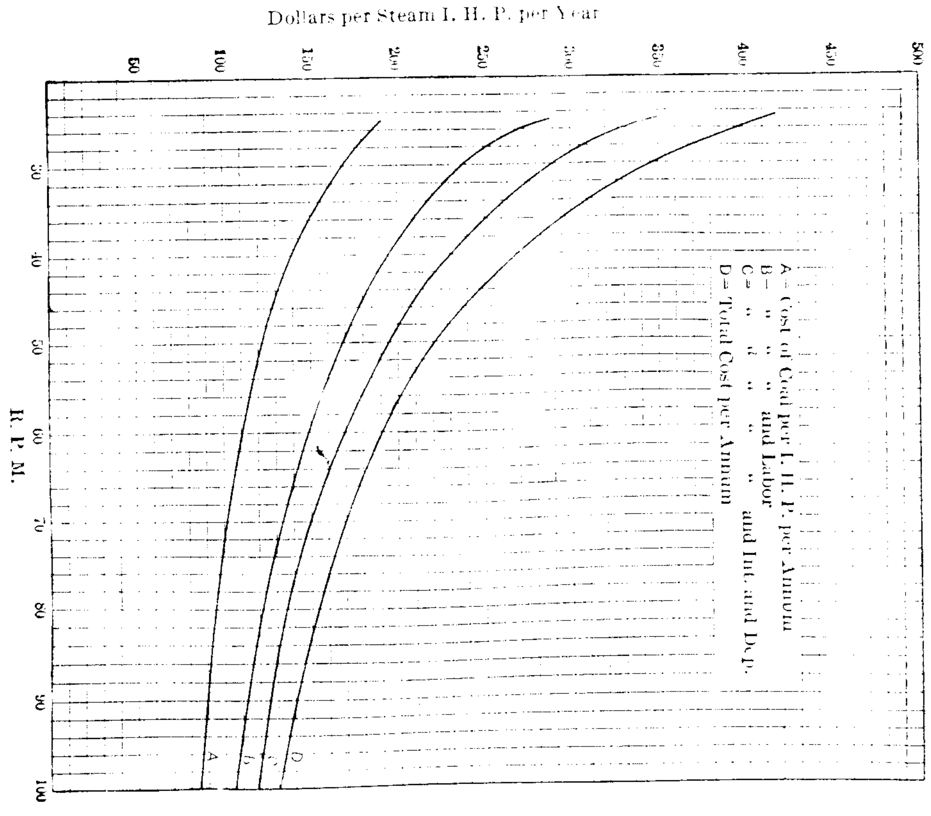

FIG. 11.—Compressor Plant No. 1.

compressor 97.81 ÷ 161 = 60.8 per cent. Table VI shows the cost of running this compressor at different speeds for a period of one year. Fig. 11 shows how the cost per steam horsepower per year is affected by the average running speed.

Plant Two consisted of three 150-hp. boilers, supplying steam for a Corliss engine, a duplex compressor, and steam heating. To find the

TABLE VI.—TEST ON PLANT NUMBER ONE

Power Costs. Duplex Meyer Steam, Compound Air End. Operated Condensing

Rev. per min.	Evaporation	Water rate	Pounds coal per I.Hp. hour	Steam I.Hp.	Pounds coal per hour	Coal		Labor per I.Hp.	Coal and labor per I.Hp.	Int. and depreciation per I.Hp.	Coal labor, int. and depreciation per I.Hp.	Repairs, etc., per I.Hp.	Total cost per I.Hp.	Cu.ft. compressed air per hour	Operating hourly costs	Cents per 1,000 cu.ft.
						Cost per annum	Cost per I.Hp. per annum									
25	8.08	59	7.30	55	401.5	$10,550	$192.00	$97.00	$289.00	$62.00	$351.00	$62.00	$414.00	2,963	$2.59	.874
40	8.31	44	5.30	95	503.5	13,240	130.00	56.00	195.00	36.00	231.00	36.00	268.00	4,238	2.90	.684
60	8.62	36.5	4.23	152	642.9	16,900	111.00	35.00	146.00	23.00	169.00	23.00	191.00	5,963	3.32	.557
80	8.82	32	3.67	211	774.4	20,350	96.45	25.00	121.00	16.00	137.00	16.00	154.00	7,650	3.72	.487
100	8.96	29.5	3.29	278	914.6	24,050	86.60	19.00	106.00	12.00	118.00	12.00	130.00	9,308	4.14	.445

TABLE VII.—TEST ON PLANT NUMBER TWO

Duplex Simple Steam Cylinders, 14 in. × 22 in., Meyer Valve Gear; Two-stage Air End, 22 in. and 14 in.; 22 in. Stroke. Rated Capacity, 1,050 Cu.ft. per Minute at 105 Rev. per Min.

Rev. per min.	Gage pressure, lb.		High-pressure Side				Low-pressure Side				Steam total, I.Hp.	Air total, I.Hp.	Per cent air of steam	Pounds steam, per I.Hp. hour	Metered air output, cu.ft. per min. 70° F.	Displacement, cu.ft. per min. 70° F.
			Steam		Air		Steam		Air							
	Steam	Air	M.E.P.	I.Hp.	M.E.P.	I.Hp.	M.E.P.	I.Hp.	M.E.P.	I.Hp.						
14.3	90	190	44.62	11.3	43.8	10.7	43.2	10.55	14.4	8.71	21.85	19.4	88.9	78.9	295	151
46.0	90	95	45.5	37.0	41.1	32.4	43.3	34.0	15.75	30.6	71.0	63.0	88.8	46.32	564	484
51.0	85	98	45.4	39.6	41.6	36.3	43.9	38.2	15.06	33.4	77.3	69.7	90.3	53.9	654	535
82.0	85	55	42.2	59.1	25.6	35.9	32.5	45.5	15.8	54.6	104.6	90.5	86.5	42.4	744	864
103.0	77	93	48.9	86.1	40.1	70.5	44.1	78.2	17.03	74.0	164.3	144.5	88.0	43.1	1,000	1,083
111.0	82	75	46.7	88.6	32.3	61.4	40.5	76.9	16.8	78.5	165.5	139.9	84.6	38.9	1,025	1,161

NOTE.—The above data are from a table of readings taken at fourteen different speeds.

boiler horsepower, a meter was placed on the compressor steam pipe, so that only the steam actually used by the compressor was so charged. The compressor had simple steam cylinders 14 by 22 in., and two-stage air end, 14 and 22 in. by 22-in. stroke, rated by the maker at 1,050 cu ft. of free air per minute at 105 rev. per min. The test lasted over a month.

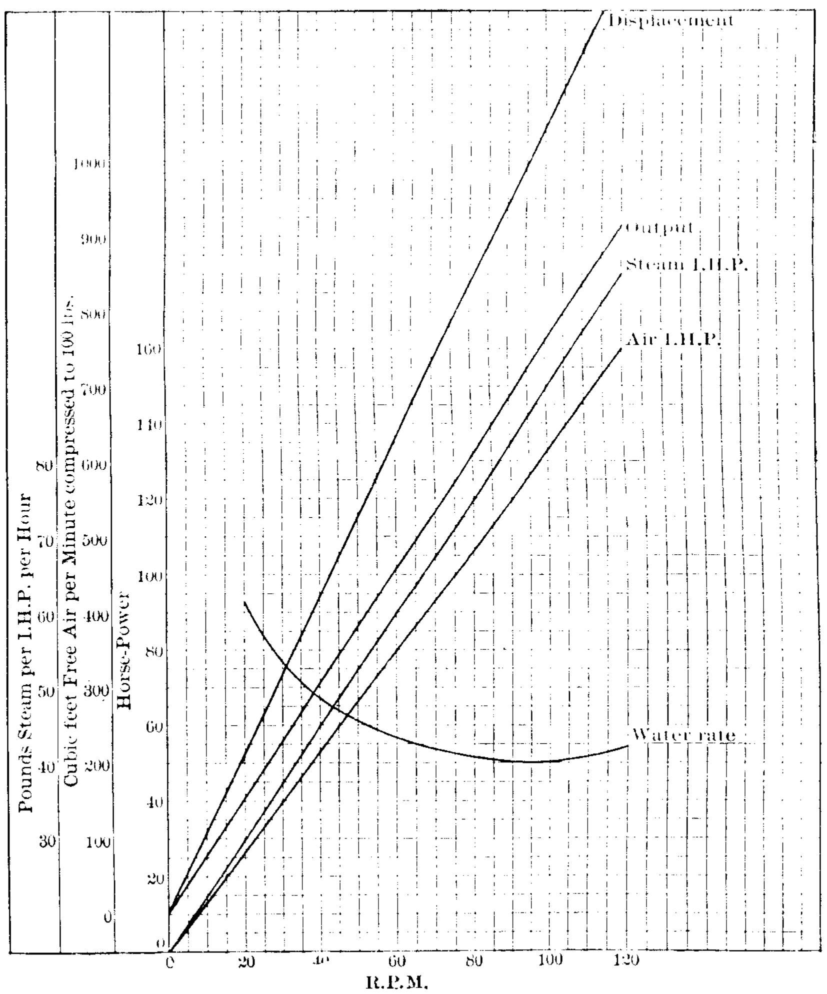

FIG. 12.—Compressor Plant No. 2.

Total coal consumed, pounds	159,250
Total feed water, pounds	2,496,000
Average evaporation per pound coal consumed, pounds	5.46
Average revolutions per minute	36
Corresponding average indicated horsepower (from curve)	53

Hourly readings of the counter gave an average speed of 36.05 rev. per min., at which the steam consumption at the throttle was 51 lb. per I.Hp.-hour, the air meter showing a delivery of 275 cu.ft. of free air per minute. The total efficiency was 67 per cent. By the ordinary method of computing the mechanical efficiency only, at the same speed there would be 48 air hp., divided by 54 steam hp., giving an efficiency of 89 per cent.

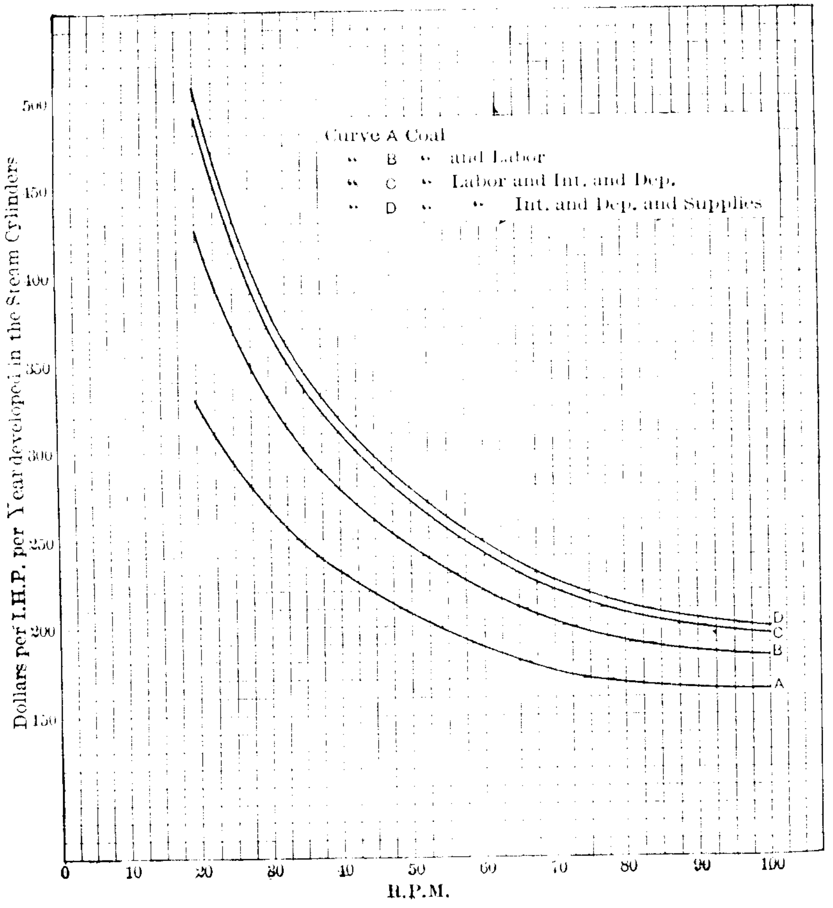

FIG. 13.—Compressor Plant No. 2.

The coal consumption per indicated horsepower per year, at the average speed, as shown by the company's books, was about 56 tons. Table VII and Figs. 12 and 13 present details of this test.

Plant Three comprised two 125-hp. tubular boilers, supplying steam to a non-condensing cross-compound compressor; steam cylinders 18 and 35 in.; air cylinders 14 and 28 in. by 24-in. stroke. A two weeks' run gave:

Total coal burned, pounds	221,190
Total feed-water, cubic feet	34,273
Total feed-water, pounds	2,094,657
Average temperature feed-water, degrees F.	154
Average evaporation per pound coal consumed, pounds	9.48
Average boiler horsepower	208
Average revolutions per minute	66
Average indicated horsepower of steam end, at 66 revolutions per minute (from curve)	240
Average indicated horsepower of air end (from curve)	128.5
Average steam pressure	97
Average air pressure	97
Average outside temperature, degrees F.	23
Average air piston displacement at normal speed, cubic feet, at 70° F.	1,372
Metered output in cubic feet corrected to 70° F.	734

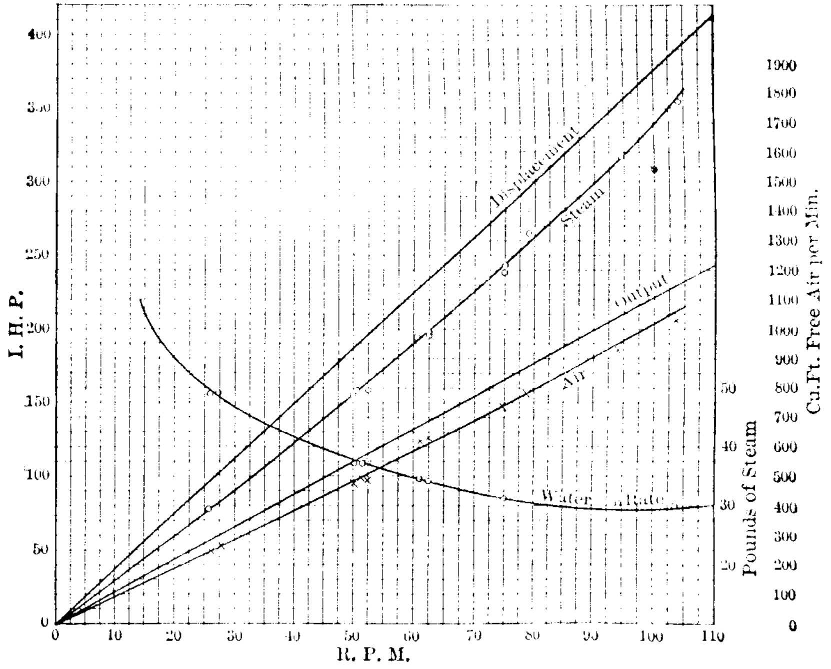

FIG. 14.—Compressor Plant No. 3.

The average evaporation of 9.48 lb. water per pound of coal, from 154° F. to 97 lb. average steam pressure, is equivalent to 10.4 lb. of water evaporated from and at 212° F. At 66 rev. per min., the displacement was 1,240 cu.ft. of free air per minute, and the metered output was 734 cu.ft., showing a net volumetric efficiency of 59 per cent.

Table VIII.—Test on Plant Number Three

Compound Non-condensing Steam Cylinders, 18 in. and 35 in.; Two-Stage Air End, 28 in. and 14 in. by 24 in. Stroke

Rev. per min.	Steam pressure, lb.	Air pressure, lb.	High-pressure Side				Low-pressure Side				Total steam, I.Hp.	Total air, I.Hp.	I.Hp. friction	Per cent loss in friction	Air line temperature	Cu.ft. displacement, 70° F.	Output, cu.ft. air per min., 70° F.
			Steam		Air		Steam		Air								
			M.E.P.	I.Hp.	M.E.P.	I.Hp.	M.E.P.	I.Hp.	M.E.P.	I.Hp.							
24	100	75	53.6	38.6	35.7	15.6	8.93	24.8	13.85	24.7	63.4	40.3	23.1	36.4	120	447	269
29.6	98	67	50.55	44.9	28.5	15.4	7.32	25.0	10.43	29.5	69.9	44.9	25.0	35.8	119	550	330
50	97	100	61.6	92.4	45.55	41.5	10.67	61.7	14.48	53.7	154.1	95.2	52.9	38.2	120	935	555
61	100	102	60.6	110.9	45.8	50.9	11.6	81.8	15.1	68.4	102.7	119.3	73.4	38.1	140	1,145	677
75	99	100	62.6	140.9	44.68	61.0	11.76	102.0	15.36	85.5	242.9	146.5	96.4	39.7	160	1,409	831
94.5	95	99	48.7	138.0	44.95	77.4	16.31	178.3	15.64	109.7	316.3	187.1	129.2	40.85	165	1,780	1,050
103.5	98	95	47.7	148	43.0	81.1	17.2	206	16.31	125.3	334.0	206.4	147.6	45.92	173	1,948	1,150

Note.—The above data are from a table of eighteen different readings taken at various speeds.

To determine the average operating conditions, the curve at 66 rev. per min. is followed (Fig. 14), at which the indicated horsepower of the steam cylinders was 210, and that of the air cylinders, 128. This shows the efficiency to be 61 per cent, the friction loss being 81.5

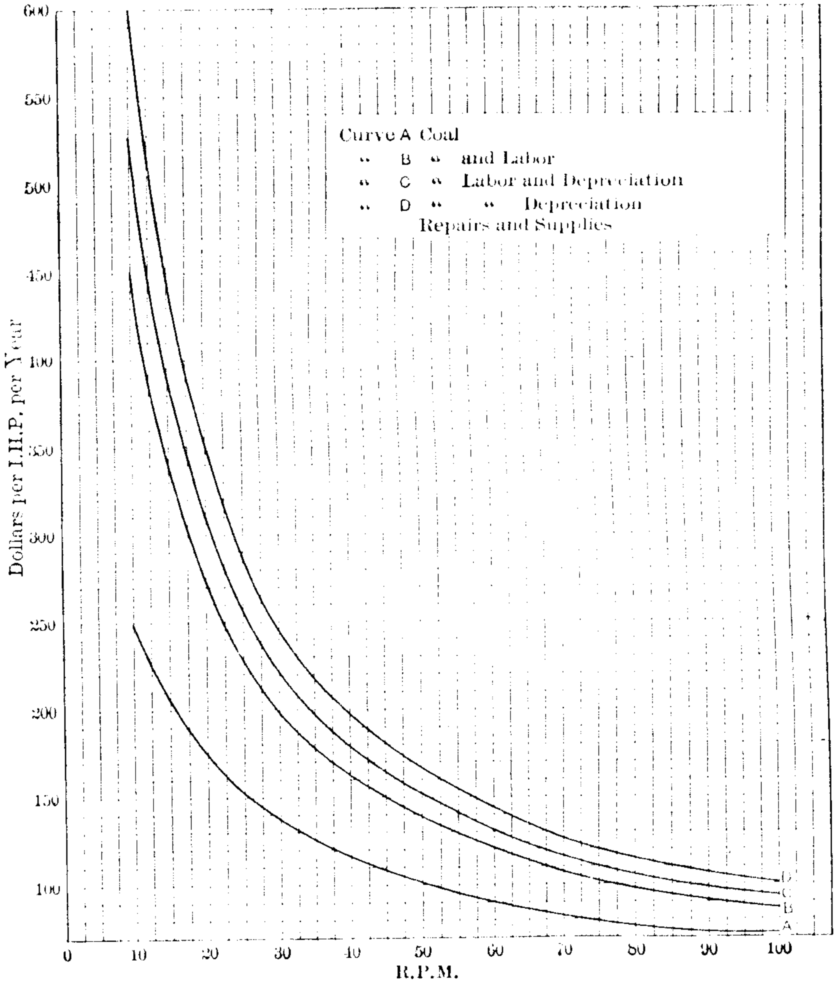

Fig. 15.—Compressor Plant No. 3.

Hp., or 39 per cent of that delivered by steam end. This extremely high friction loss was caused by the fact that the compressor shaft was out of line, and the plant could not be shut down long enough to rectify it. The details of the test, given in Table VIII and Figs. 14 and 15 emphasize the inefficiency that may be due to a purely mechanical defect.

Summary.—The results of these tests reveal the amount of the losses occurring when a compressor operates under the unfavorable conditions of varying air consumption, unavoidable where machine drills use most of the air, and emphasize the importance of making tests at reasonable intervals.

Again, compressors generally develop less than their full rated power. A compressor is essentially a variable-speed machine, its speed being controlled according to the air demand by some form of governor (see Chapter VIII). It would be well if all compressor builders gave in their catalogs the horsepower rating and cost per air horsepower delivered at different speeds, together with efficiencies at different loads and speeds.

CHAPTER XI

COST OF COMPRESSORS AND COMPRESSED AIR

Early in the period of the World War, rates of wages and costs of materials began to rise; but, in the United States, it was not until about January, 1916, that this increase became serious. From that time until the middle of 1920, twenty months after the Armistice, costs of commodities rose rapidly to a rough maximum of 220 per cent of pre-war costs (120 per cent above 1914 costs taken as 100). A rapid drop then ensued, to about 170 per cent of pre-war costs in the spring of 1921. Since then average prices have fluctuated within the relatively narrow limits of 165 to 180 per cent of 1914 costs, and are now (October, 1929) but little lower than in April, 1921. It seems probable that costs will be maintained roughly at this level for some time to come.

Average rates of wages, as reflected by earnings of factory and mill operatives, kept fairly close pace with the rise of commodity costs until early in 1920. But, in the fall of that year, earnings rose to a higher maximum—230 per cent of pre-war rates—followed by a drop to 190 per cent at the end of 1921. Finally, during the period of relatively small fluctuations in commodity costs, average earnings continued to rise, with minor changes, until, in October, 1929, they reached about 235 per cent of pre-war rates.* Minor variations in these costs from 1914 to 1928 are exhibited by Fig. 1; the curve of commodity costs represents the general " cost of living " in the United States; the other curves, showing earnings, apply in particular to the state of New York.

The trend of costs during the past fifteen years, as outlined, will aid in comprehending some of the economic changes wrought by the World War. It is believed that this résumé will be useful, especially to the younger engineers whose experience does not reach back to the pre-war period and who therefore may not fully realize the wide differences between former and present prices.

Preliminary Estimates.—The specific purpose in recording here the above changes is to furnish data for making approximate estimates of the cost of compressor plants. In the great mass of pre-war engineering literature, there are many detailed statements of first costs and operating expenses (some of which are cited hereafter in this chapter) that can still be used as a basis for rough estimates, provided a proper percentage increase is applied to bring such costs up to present-day levels. Estimates thus made can be checked by the figures in Table I, which gives the approximate unit costs, at the close of the year 1929, of standard compressors of different types, expressed in terms of the number of cubic feet of free air compressed and delivered per minute. This is a convenient basis, commonly adopted by both builders and users of compressors.

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* Reports of the Federal Departments of Commerce and of Labor.

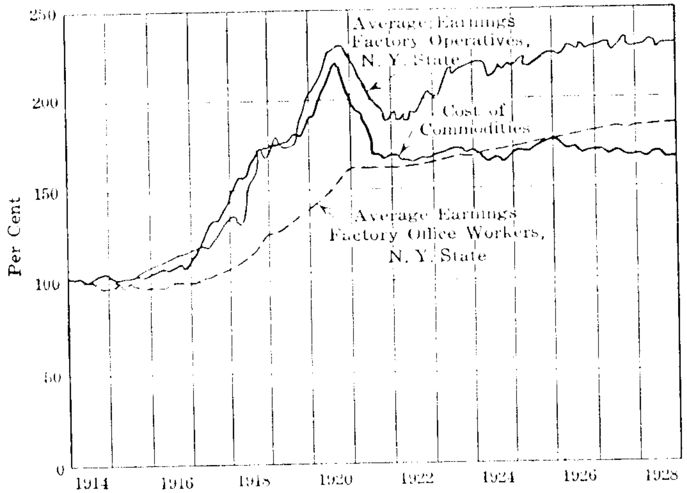

FIG. 1. Average Costs of Commodities, and Earnings of Factory and Mill Operatives in New York State, from 1914 to 1928. (*Bulletin*, Federal Reserve Bank.)

The types of compressor listed in Table I are made by most of the builders, and, with the figures designating their capacity, represent fairly the range of current practice. Illustrations and specifications of these types are contained in Chapter III. All accessories recognized as being necessary features of standard equipment are covered by the stated costs. Other items, such as special forms of governors, regulators, unloaders, automatic water valves, intake screens, etc., are not included.

Fig. 2 will also be useful in making preliminary estimates. The curves show initial pre-war costs in Great Britain (£ sterling) of steam-driven two-stage compressors of 100 to 1,500 hp., and the comparative costs of the same plants computed at from $1\frac{1}{2}$ to 3 times the pre-war figures. These ratios are obviously of general application.

TABLE I. COST OF COMPRESSORS PER CU.FT. OF FREE AIR PER MIN., COMPRESSED TO 100 LB. (Without accessories other than those of standard equipment)

Type of compressor	Capacity range, cu.ft. free air per min.	Cost per cu.ft. of free air
Straight-line, steam-driven	67-386	$15.90-$6.80
Straight-line, belt-driven	11-528	6.25-2.85
Vertical, single-cylinder, belt-driven	7.3-44	7.85-4.50
Vertical duplex, direct electric-driven	58-316	11.20-6.80
Cross-compound, two-stage air, duplex steam, Meyer valve	246-1,723	11.10-5.95
Cross-compound, two-stage air, duplex steam, piston valve	246-3,026	11.10-5.75
Cross-compound, two-stage, direct electric-driven	310-3,864	9.50-4.90
Cross-compound, two-stage, belt-driven	318-1,575	5.00-3.55
Horizontal, single-acting, two-stage, driven by horizontal oil engine	603-895	15.10-14.80
Vertical, single-acting, two-stage, driven by horizontal oil engine	354	15.50
Horizontal, double-acting, duplex or two-stage, driven by vertical, duplex, single-acting, 4-cycle gas engine*	143-1,938	34.00-31.00
Portable, truck-mounted, vertical, duplex, gasoline-driven	66-310	15.90-11.30
Mine-car type, vertical, duplex, electric-driven	70-250	18.00-12.00

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* The high first cost of compressors driven by gas engines is usually offset in part by the relative cheapness of fuel gas.

Data Required by Compressor Builders for Furnishing Estimates. — It will be understood that what precedes refers only to rough estimates, as usually made by an operating manager or engineer, on the basis of ascertained local conditions. To obtain exact costs of compressors for a given service, estimates or bids are requested from the makers, to whom must be submitted as complete a statement as possible of the operating conditions. The following outline will serve as a general indication of the points to be covered in applying to makers:

1. Purpose for which the compressed air is to be used.
2. Working air pressure required; also volume of free air to be supplied, in cubic feet per minute.
3. Altitude above sea-level at which the compressor is to operate.
4. Type of compressor required or preferred, and whether portable or stationary (to be erected on the surface or underground).

5. If the compressor is to be steam-driven, state steam pressure available, kind and cost of fuel, and whether engine is to be condensing or non-condensing.

6. If for driving by electric motor, state whether direct-connection or belt-drive is preferred; also details as to electric current available.

7. If for belt-drive, state horsepower at belt and maximum distance allowable between driving centers.

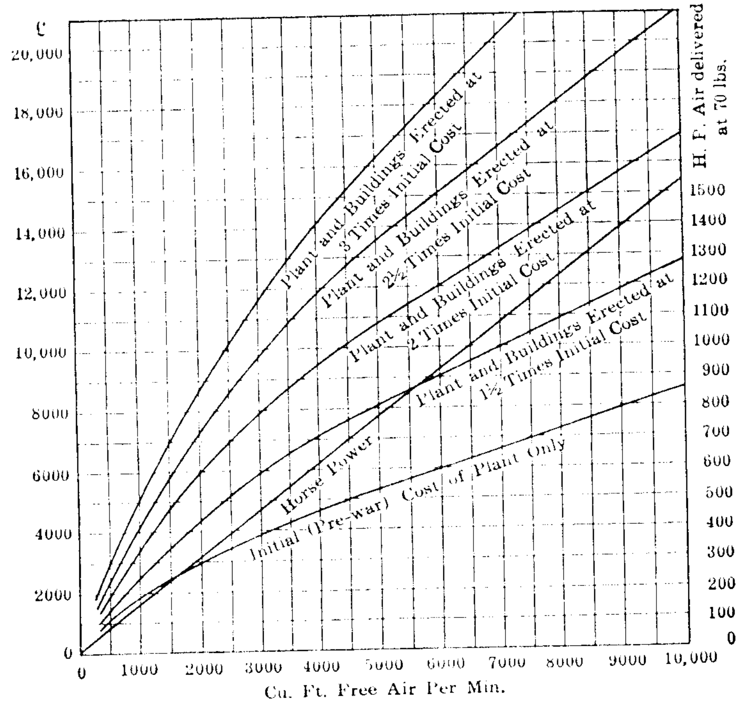

FIG. 2.—Approximate Cost of Compressed-air Installations; Two-stage, with Compound, Non-condensing Steam Cylinders; Air Pressure, 70 Lb. (Daw, *Compressed Air Power.*)

8. If water power is to be used, state head of water and cubic feet of water per minute.

9. State whether the demand for air will be constant or intermittent.

10. State kind, number and size of machine drills to be operated.

11. If the air is to be used for pumps, hoists or compressed-air locomotives, give their make, size, and speed or capacity; also, for pumps, the head of water.

12. State transport facilities; if compressor must be sectionalized, give the weight allowable for heaviest part or package.

Cost of Installation.—It has been shown that the first cost of compressors can be obtained directly from makers, or estimated in advance with reasonable accuracy. The cost of installation, on the other hand, is a wholly different matter. It is so variable, in its dependence upon the type of compressor, prices of materials and local rates of wages, that it is difficult to make a close estimate. Since it is obvious that this cost can be known exactly only after the plant has been erected, an advance estimate must usually be based upon the costs of similar plants that have been installed under similar local conditions.

It is sometimes suggested that the cost of the foundation and erection of a compressor or other engine can be computed roughly as a percentage of the first cost—say 15 or 20 per cent. But, even when this is done, the only mode of fixing upon the proper percentage to use is to ascertain what these costs have been in the case of other plants of the same general type. Therefore, in preparing for such estimates, the best the engineer can do is to assemble and compare whatever data may have been published or are otherwise available. The examples from practice cited on subsequent pages will be of service in this connection.

Maintenance and Running Expenses.—Preliminary estimates of these costs can be made only by applying the same considerations as were outlined above respecting cost of installation. Their determination must generally depend upon analogy, combined with the judgment and experience of the engineer.

Cost of Compressing Air by a plant of given type can be estimated with considerable accuracy. Certain factors entering into the problem have been compiled by engineers and compressor builders.

Take, for example, a compressor of ordinary size, say of approximately 100 hp. It may be direct-connected to, or belt-driven by, a steam or gasoline engine or an electric motor. The following tables show the comparative costs of compressing and delivering 100 cu.ft. of free air, at a gage pressure of 90 lb., for each of these types of plant. In using the tables, after first finding the horsepower (see Chapter X), a proper comparison between the results can be made by assuming certain figures in accordance with local conditions and the accepted principles of practice; the same mechanical efficiency being taken for all the types of plant cited.

In computing comparative costs, as above, it is obvious that the element of time occupied in compressing the assumed unit volume of 100 cu.ft. of air needs not be considered.

The figures given in Table II are standard for steam plants; percentage variations from them must appear in the calculation for a given case. The figures for cost of the coal consumed, which are based on a price of $1.00 per ton for steam coal, are readily changed for any other market price.

TABLE II.—COST OF COAL TO COMPRESS AND DELIVER 100 CU.FT. OF FREE AIR (H. V. Conrad)

Cost of coal taken at $1.00 per ton of 2,000 lb.

Brake hp. for 100 cu.ft. free air per minute										
	Pounds of steam required by engine per horsepower-hour									
	24	26	28	30	32	34	36	38	40	42
	Coal per lb. steam, when boiler evaporates 7 lb. water per pound of coal									
	3.43	3.714	4.00	4.286	4.571	4.757	5.143	5.428	5.714	6.00
	Coal cost per hp.-hr. at $1.00 per ton, fractions of cent									
	.1715	.1857	.200	.2143	.2285	.2378	.2571	.2714	.2857	.300
	Coal cost per 100 cu.ft. free air, fractions of cent									
16	0.0457	0.0495	0.0533	0.0572	0.0610	0.0634	0.0686	0.0724	0.0762	0.0800
18	.0514	.0557	.0600	.0643	.0686	.0714	.0772	.0814	.0857	.0900
20	.0570	.0619	.0667	.0714	.0762	.0793	.0857	.0904	.0952	.1000
22	.0627	.0681	.0733	.0785	.0838	.0872	.0943	.0995	.1047	.1100
24	.0684	.0743	.0800	.0857	.0914	.0951	.1029	.1085	.1142	.1200
26	.0741	.0805	.0866	.0929	.0990	.1030	.1114	.1175	.1238	.1300
28	.0798	.0867	.0933	.1000	.1066	.1109	.1200	.1266	.1334	.1400
30	.0855	.0929	.1000	.1071	.1142	.1189	.1285	.1357	.1429	.1500

Example.—If 22 brake hp. are required to deliver 100 cu.ft. free air per minute at 90 lb. gage, and the engine uses 34 lb. steam per horsepower-hour, and the boiler evaporates 7 lb. of water per pound of coal; then, with coal at $6.00 per ton, its cost per 100 cu.ft. of air is 0.0872 × 6 = 0.5232 cent.

In comparing the cost of compressed air, delivered as above by a steam-driven compressor, with the cost when the compressor is driven by any other source of power, the expense of operating the boiler plant is also a factor.

Steam Consumption of Compressors, per unit volume of air compressed, depends upon the delivery pressure, the type of the steam end and the number of stages. The weights of steam required to compress 100 cu.ft. of free air adiabatically to different gage pressures, by single and two-stage compression, as given in Tables III and IV (O. S. Shantz, *Power*, Feb. 4, 1908, and Ingersoll-Rand Co., "Compressed Air Data") are useful for cost computations. For two-stage compression, it is assumed that the air is cooled between the stages to atmospheric temperature, and that the mechanical efficiency of the compressor is 90 per cent (see Chapter X, under "Efficiency"). The figures for intermediate rates of steam consumption not shown in the Tables are found by interpolation.

In using Tables III and IV, the steam consumption per indicated horsepower for any given case is assumed in accordance with good practice, to suit the type of the steam end, steam pressure, cutoff, vacuum if condensing, and size of compressor. Obviously, therefore, the accuracy of the tables will depend upon the correct assumption of the rate of the steam consumed. But, even in cases where this rate can not be exactly determined, comparisons can be made of the fuel required for different types of compressor, to show the approximate saving by using compound instead of simple steam cylinders.

Examples.—I. A straight-line, single-stage compressor, using 30 lb. of steam per indicated horsepower for compressing to 100 lb., required 9.9 lb. steam per 100 cu.ft. of air; while a duplex, two-stage compressor, at the same steam consumption, required 8.42 lb. steam. II. A two-stage, compound, non-condensing compressor, with a 26-lb. steam rating, required 7.3 lb. steam per 100 cu.ft. of air; while a large Corliss, two-stage, compound, condensing compressor, using high-pressure steam, and having a water rate of 17 lb., including the condenser, required only 4.77 lb. steam.

The performance of compressors can also be compared on the basis of pounds of coal consumed per indicated horsepower, instead of pounds of steam. This is done by dividing the steam consumption in the tables by 7, since a fair average evaporation is 7 lb. water per pound of coal. Knowing the local price of coal, the comparison is thus stated in terms of dollars and cents.

Steam consumption can be computed roughly from the indicator card of a compressor. A considerable part of the steam supplied is condensed on entering the cylinder, and is not re-evaporated until after the exhaust valve opens. As this steam therefore does not show on the indicator diagram, the results are only approximate.

TABLE III.—POUNDS OF STEAM TO COMPRESS ADIABATICALLY 100 CU.FT. OF FREE AIR BY SINGLE-STAGE COMPRESSION

Air pressure, pounds	Steam per indicated horsepower per hour, pounds										
	16	18	20	22	24	26	28	30	32	36	40
20	1.82	2.04	2.26	2.49	2.72	2.94	3.17	3.40	3.61	4.08	4.54
30	2.45	2.76	3.06	3.37	3.68	3.98	4.29	4.60	4.90	5.51	6.12
40	3.02	3.39	3.77	4.15	4.52	4.90	5.26	5.65	6.03	6.78	7.50
50	3.50	3.93	4.36	4.80	5.25	5.68	6.10	6.55	7.00	7.86	8.71
60	3.90	4.38	4.80	5.36	5.85	6.32	6.80	7.30	7.80	8.76	9.71
70	4.30	4.83	5.36	5.90	6.45	6.97	7.50	8.05	8.60	9.66	10.70
80	4.67	5.25	5.84	6.42	7.00	7.59	8.15	8.75	9.34	10.50	11.61
90	4.96	5.58	6.20	6.82	7.45	8.05	8.66	9.30	9.94	11.15	12.35
100	5.29	5.95	6.60	7.25	7.92	8.58	9.22	9.90	10.56	11.88	13.15
110	5.56	6.26	6.96	7.66	8.36	9.05	9.75	10.45	11.15	12.52	13.90
120	5.82	6.57	7.30	8.04	8.76	9.50	10.20	10.95	11.66	13.13	14.55

TABLE IV.—POUNDS OF STEAM TO COMPRESS ADIABATICALLY 100 CU.FT. OF FREE AIR BY TWO-STAGE COMPRESSION

Air pressure, pounds	Steam per indicated horsepower per hour, pounds										
	16	18	20	22	24	26	28	30	32	36	40
70	3.76	4.23	4.69	5.16	5.63	6.10	6.56	7.04	7.50	8.45	9.35
80	4.03	4.52	5.02	5.53	6.03	6.53	7.03	7.53	8.03	9.05	10.01
90	4.26	4.79	5.32	5.85	6.38	6.91	7.44	7.95	8.50	9.57	10.60
100	4.50	5.05	5.61	6.19	6.74	7.30	7.85	8.42	8.99	10.10	11.20
110	4.74	5.32	5.91	6.51	7.10	7.70	8.27	8.86	9.46	10.64	11.80
120	4.93	5.54	6.15	6.78	7.38	8.00	8.61	9.24	9.85	11.05	12.27
130	5.11	5.75	6.38	7.03	7.66	8.30	8.92	9.57	10.20	11.45	12.72
140	5.29	5.94	6.60	7.26	7.92	8.60	9.23	9.90	10.56	11.88	13.15
150	5.46	6.14	6.81	7.50	8.17	8.86	9.55	10.20	10.90	12.26	13.60

TABLE V.—COST OF GASOLINE IN CENTS TO COMPRESS AND DELIVER 100 CU.FT. OF FREE AIR (Ingersoll Rand Co.)

Gasoline Consumption, 1 Pint per B.Hp. per Hour

Brake hp. to deliver 100 cu.ft. free air per minute	Price of gasoline per gallon, cents							
	8	9	10	11	12	13	14	15
16	0.266	0.300	0.333	0.366	0.400	0.433	0.466	0.500
18	.300	.337	.375	.413	.450	.487	.525	.562
20	.333	.375	.416	.458	.500	.542	.584	.626
22	.366	.412	.458	.504	.550	.595	.642	.687
24	.400	.450	.500	.550	.600	.650	.700	.750
26	.433	.488	.542	.599	.650	.705	.759	.813
28	.466	.525	.583	.642	.700	.758	.816	.875
30	.500	.563	.625	.688	.750	.813	.875	.938

Though the gasoline consumption per brake horsepower will not vary greatly, the percentage of variation should be considered.

Example.—If the price of gasoline is 20 cents per gallon, and 22 b.hp. are required to deliver 100 cu.ft. free air per minute, at 90 lb. gage, the gasoline cost for this volume of air will be 0.458 × 2 = 0.916 cent.

TABLE VI.—COST OF ELECTRIC CURRENT IN CENTS TO COMPRESS AND DELIVER 100 CU.FT. OF FREE AIR (Ingersoll-Rand Co.)

Brake hp. to deliver 100 cu.ft. free air per minute	Price of electric current per kilowatt-hour, cents								
	1.	1.5	2.	2.5	3.	3.5	4.	4.5	5.
16	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0
18	.225	.337	.45	.562	.675	.787	.9	1.012	1.125
20	.25	.375	.5	.625	.75	.875	1.0	1.125	1.25
22	.275	.412	.55	.687	.825	.962	1.1	1.237	1.375
24	.3	.45	.6	.75	.9	1.05	1.2	1.35	1.5
26	.325	.487	.65	.812	.975	1.137	1.3	1.46	1.625
28	.35	.525	.7	.875	1.05	1.225	1.4	1.575	1.75
30	.375	.562	.75	.937	1.125	1.312	1.5	1.687	1.875

Example.—If electric current costs 4 cents per kw.-hr., and 22 b.hp. are required to deliver 100 cu.ft. free air per minute at 90 lb. gage, the cost of electric current for this volume of air will be 1.1 cent.

In using Table VI it is assumed that electric current is purchased from some power plant at a contract price. The cost of locally generated electricity is generally higher.

Saving Due to Compressor Regulation.—In selecting a compressor for given service, its capacity should be equal to the maximum demand for air. Variations in the demand are taken care of by some form of governor, regulator or unloader (or a combination of them), the current designs of which are dealt with in Chapter VIII. The power input, being thus adjusted to the load factor (Chapter X), a saving in cost of operation is effected.

Furthermore, where the variations in load are likely to be large, it is advisable to divide the total compressor capacity between several units, one or more being kept running at full load. This aids in maintaining maximum economy, regulation then depending upon automatic starting and stopping of the several units. Step-regulation is especially advantageous for compressors driven by constant-speed, direct-connected electric motors.

The effect of variations in output upon the horsepower developed by a compressor is illustrated by the indicator cards in Fig. 22, Chapter VIII, which show clearly the large percentage of saving that can be realized by proper regulation.

Cost of Leaks in compressed air plants is a matter deserving more attention than it usually receives. In its handbook of *Compressed Air Data*, the Ingersoll-Rand Co. prints the significant Table VII, from a sign conspicuously posted in the power plants of the Rock Island Railroad. The figures in this table show the importance of stopping even small leaks of compressed air. New gaskets for piping, etc., are cheap, and periodical inspection of compressor plants is well worth while. For the use of air meters for measuring wastage due to leaks, see *Engineering and Mining Journal*, Aug. 25, 1923, p. 331.

TABLE VII.—COST OF AIR, STEAM AND WATER LEAKS

Size of opening, in.	Air		Steam		Water	
	Cu.ft. wasted per month, 100 lb. pressure	Cost of waste per month, at 10c per 1,000 cu.ft.	Pounds wasted per month, 100 lb. pressure	Cost of waste per month, at 60c per 1,000 lb.	Gallons wasted per month, 40 lb. pressure	Cost of waste per month, at 15½c per 1,000 gal.
3/8	9,979,200	$997.92	460,000	$276.00	692,400	$108.00
1/4	4,449,600	444.96	203,000	121.80	307,700	48.00
1/8	1,114,560	111.46	50,500	30.30	76,900	12.00
1/16	278,640	27.86	12,750	7.65	19,200	3.00
1/32	69,552	6.96	3,175	1.91	4,800	0.80

EXAMPLES OF COSTS

The following data can be used in making preliminary estimates, by applying the percentage increases of present-day costs, as set forth at the beginning of this chapter.

Cost of a compressor plant complete for drills and pumps in driving the siphon tunnel under the Hudson River, Storm King, N. Y., in 1908. It comprised: two class HH-3 Ingersoll-Rand cross-compound condensing compressors, 16 by 28 by 25¼ by 16¼ by 16, each with a displacement of 1,392 cu.ft. of free air compressed to 100 to 110 lb.; steam pressure, 150 lb.; one 48-in. aftercooler and one 54-in. by 12-ft. air receiver; three 130-hp. Sterling boilers; two 6 by 4 by 6 boiler feed pumps; two 6 × 5.75 × 6 tank pumps; one 10 by 18 by 10 independent jet condenser; one 400-hp. Berriman feed-water heater; one 20-kw. Kerr steam-turbine generating set; one station panel with switches; one feed-water tank; 2500 ft. of 6-in. W.-I. pipe; and 2500 ft. of 1.5-in. 2-conductor cable. Total cost, including railroad siding, building and foundation, piping and power house, boiler setting, and all labor and accessories, was about $35,000. Of this, the compressors, aftercooler and receiver cost $13,500, and the boiler, pumps, generator set, water tank, piping, and electrical conductor, about $10,000. The railroad siding, building and foundations, piping and power house, boiler settings, well, stacks, labor, superintendent's charges for placing plant in operation, and incidentals, cost $11,500.

Fig. 3 shows the costs of compressing air at different load factors by the following electric-driven compressors (W. C. Lancaster, *Power*, Feb. 3, 1914):

	COMPRESSOR A	COMPRESSOR B
Dimensions, inches	26 × 15½ × 18	22 × 14 × 18
Drive	Direct-connected	Belted
Revolutions per minute	185.3	157.8
Displacement, cubic feet per minute	2,039.7	1,241.2
Discharge pressure, pounds	100	100
Volumetric efficiency, per cent	86.5	88.6
Horsepower per 100 cu.ft. free air per minute	22.75	23.70

Curves 2 and 3 in Fig. 3 are for the same compressor; the former being based on a price for electric current of 1 cent per kilowatt-hour and the latter on a price of $25 per horsepower of maximum demand per year. These curves show that the 1 cent rate gives better results at low load factors, but not so good at the higher load factors.

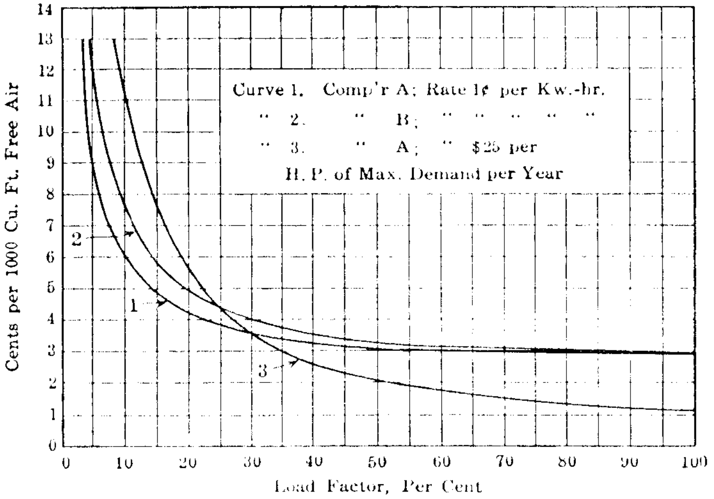

FIG. 3.—Cost of Electric Current for Compressing Air (*Power*)

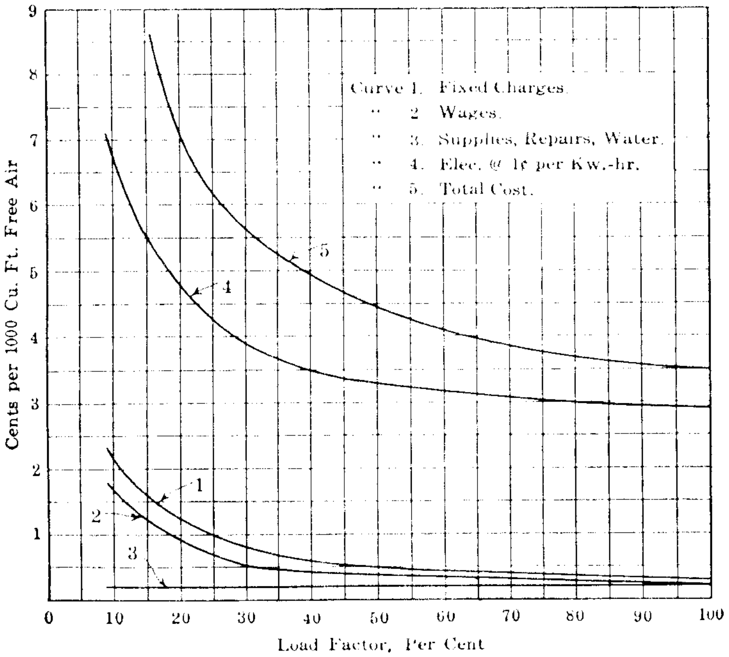

FIG. 4.—Cost of Compressing Air to 107 lb. at Different Load Factors (*Power*, Feb. 3, 1914)

Fig. 4 shows the results from a plant of four 22 × 14 × 18-in. two-stage, cross-compound compressors, belt-driven by induction motors, and one 26 × $15\frac{1}{2}$ × 18-in. compressor of the same type, direct-connected to a synchronous motor. This plant was used for a tunnel on the Canadian Northern R.R., in 1914. The compressors, made in the United States, were subject to a high duty; hence, their first cost was relatively high. As the rate paid for electric current was complicated, a flat rate of 1 cent per kilowatt-hour is assumed in computing the curves (W. C. Lancaster).

The curves in Fig. 4, based upon the power input of all the compressors, considered as a unit, were worked out from the following cost details of the five compressors:

First cost, including motors and switchboard		$31,000
Piping, aftercooler and air receiver		3,000
Building and foundations (land cost omitted)		2,700
Installation		2,300
Total first cost		$39,000
Fixed charges per year:		
Interest on $39,000 at 6 per cent	$2,340	
Taxes, 1 per cent on building and land	200	
Insurance, 20 cents per $100 per year	78	
Depreciation, 10 per cent per year	3,900	6,518
Operating costs:		
Power, based on $78\frac{1}{2}$ per cent load factor	$63,422	
Water, 7,000 cu.ft. per day at 10 cents per 100 cu.ft., or $200 per month	2,400	
Wages, $400 per month, three 8-hr. shifts	4,800	
Supplies, average per month, $103	1,236	
Repairs, average per month, $49	588	72,446
Total yearly cost		$78,964

TABLE VIII. COSTS OF COMPRESSING AIR BY STEAM AND ELECTRICITY, AT ROSSLAND GREAT WESTERN MINES, PRIOR TO 1913 *

Work performed	Steam plant	Electric plant
Average horsepower at steam cylinder of the combined compressors	730	....
Average horsepower at switchboard	....	540
Free air compressed per min. to 95 lb. gage, cu.ft.	5,432	3,319
Average horsepower to compress 100 cu.ft. free air per min.	13.4 at steam cyl.	16.3 at motor
Pounds coal consumed during test	1,038,000	....
Pounds coal consumed per horsepower-hour	1.9	....
Cost of operation:		
Fuel or current for 30-day test	$2,880.45	$1,744.26
Employes' wages	710.00	270.00
Oil, waste, etc.	147.30	73.00
Total cost for 30 days, without maintenance and depreciation	$3,737.75	$2,087.26
Fuel or current, per horsepower-month	3.95	3.23
Wages, per horsepower-month	0.97	0.50
Oil, waste, etc., per horsepower-month	0.20	0.14
Total cost per horsepower month	$5.12	$3.87
Cost per horsepower-year	$61.56	$46.44
Cost per 100,000 cu.ft. free air compressed	1.56	1.46
Cost per drill shift	1.25	1.17

* Average results of 30-day test. 80,000 cu.ft. free air taken as daily average air consumption of 3.25-in. drill. Saving in cost of electricity over steam plant, for each 100,000 cu.ft. free air = 6.9 per cent.

Pre-war operating cost per 1,000 cu.ft. of free air, of a cross-compound two-stage compressor plant in Utah; capacity, about 1,000 million cu.ft. per month. Power-house labor, 0.52 cent; repair and maintenance labor, 0.12 cent; fuel (slack coal), 9.33 lb. per 1,000 cu.ft. of air, 1.92 cents; other supplies, 0.19 cent; total, 2.75 cents. This was the best work on a six-months' run. (F. C. Merry.)

TABLE IX.—ESTIMATED COMPARATIVE COSTS OF COMPRESSING AIR (1913)

Power used is same for the three plants	Piston compressor, belt-driven from water turbine*	Piston compressor, belt-driven from electric motor	Hydraulic compressor
Investment:			
Compressor	$1,250	$1,500	$3,750
Pelton wheel complete	1,125	....	....
Electric motor, 70 hp.	....	1,200	....
Buildings, foundations, etc	450	450	....
	$2,825	$3,150	$3,750
Interest and depreciation:			
Interest on plant at 5 per cent	$141	$150	$187
Depreciation of machines	238	270	188
Depreciation of building at 3 per cent	15	15	....
Annual operating expense:			
Wages (2 shifts)	600	600	30‡
Lubricants	75	95	....
Repairs and waste	75	100	20
Electric power, 74 hp. at 0.5¢†	....	2,200	....
Total operating costs	$1,144	$3,430	$425
Output, cubic meters per minute§	7.8	10.0	10.0
Cost per 1,000 cubic meters	$0.41	$0.95	$0.12

* Efficiency turbine, 75 per cent.

† On basis of 6,000 working hours per year.

‡ Lost wages during shut-down.

§ 10 cubic meters of air at 5.1 atmospheres = 350 cu.ft. at 80 lb. pressure.

For other examples of cost of compressing air, see latter part of Chapter X.

CHAPTER XII

AIR COMPRESSION AT ALTITUDES ABOVE SEA-LEVEL

THE air-compression equations in Chapter II, and tables of performance of compressors in the first part of Chapter X, are based upon the normal sea-level atmospheric pressure of 14.7 lb. per sq.in. At altitudes, the operation of compressors is affected by the diminished density and weight of the atmosphere. This matter is of especial importance in connection with mining, because of the large number of mines situated in mountainous regions.

In considering the operating results of a compressor working at an altitude, it is necessary to distinguish between the effect of the decreased atmospheric pressure upon its volumetric efficiency, and upon the horsepower required to compress a unit volume of air.

Volumetric Efficiency of a compressor is the ratio of the volume of free air delivered to the piston displacement, and is a measure of the output of air actually available for use. It depends upon the compression ratio, $\frac{P'}{P}$ (Tables II and III, Chapter X), and the volume of cylinder clearance (Chapter V). The heat of compression produced in the cylinder increases with the compression ratio; and, as the compression ratio in turn increases with the altitude, still more heat is generated for a given delivery pressure. The clearance volume, constant in a given compressor, is also a factor, because some of the warm compressed air is trapped in the clearance spaces, and this, on re-expanding during the return stroke to the intake pressure, is equivalent to a corresponding volume of free air not taken into the cylinder. Due to these causes, the volumetric efficiency of a compressor decreases at altitudes. It serves as a basis for comparing the capacities of different compressors operating at the same altitude, or of similar compressors operating at different altitudes. For example, the volume of air delivered at 90 lb. pressure, by a single-stage compressor operating at 10,000 ft. elevation, is only 69.6 per cent of the volume it would deliver at the same pressure at sea-level (see Table 1, column 5). That is, the volumetric efficiency at 10,000 ft. is about 70 per cent of that at sea-level.

What precedes refers to single-stage compressors only. It is evident that their lower volumetric efficiency at altitudes can be compensated by using a larger compressor. In the case of a multi-stage compressor, the volumetric efficiency depends upon the compression ratio of the intake cylinder; this, in turn, being determined by the ratio of the cylinder volumes (Chapter VI). Hence, with a proper cylinder ratio, and due to the action of the intercooler, the volumetric efficiency of a stage compressor is practically unaffected by increase in altitude.

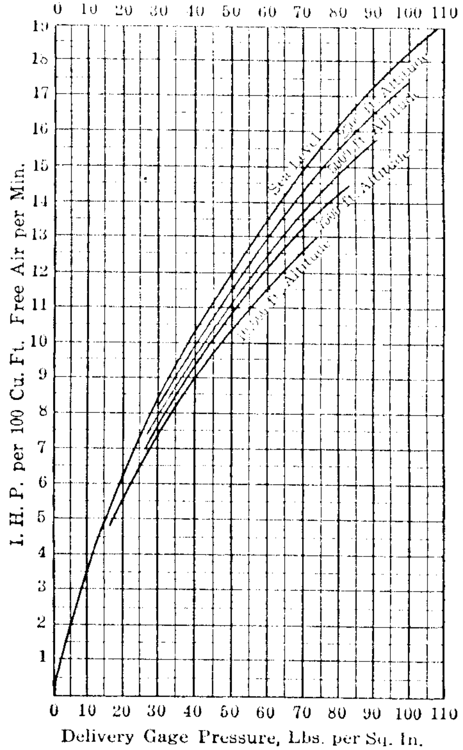

FIG. 1.—Theoretical Power for Single-stage Compression at Altitudes, Adiabatic Basis (S. B. Redfield).

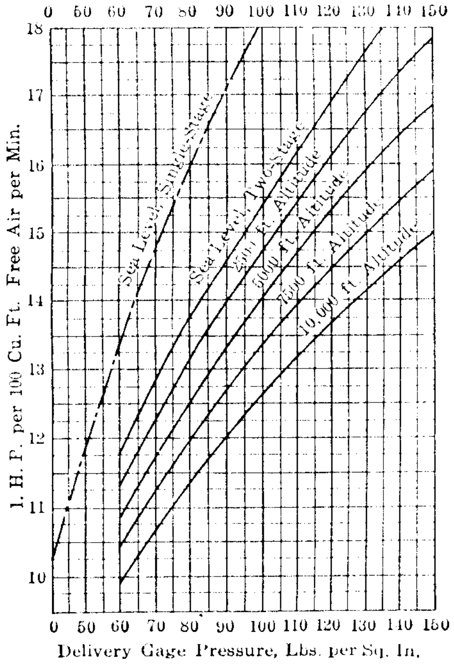

FIG. 2.—Theoretical Power for Two-stage Compression at Altitudes, Adiabatic Basis (S. B. Redfield).

Effect of Altitude upon Compressor Horsepower.—With respect to the horsepower required for both single- and multi-stage compression a fundamental distinction must be made.

1. Because of the diminished density of the atmosphere above sea-level, the horsepower for compressing a unit volume of intake *free air* to a given gage pressure decreases progressively with the altitude. Also, this horse-power is less for two-stage than for single-stage compression (Figs. 1 and 2).

II. For practical purposes it is often more useful to base the required horsepower at altitudes upon the volume of *compressed air* delivered. As shown by Table I, columns 10 and 11, the number of cubic feet of compressed air delivered per indicated horsepower decreases at altitudes; and, conversely, the horsepower per cubic foot of compressed air increases with the altitude. This increased horsepower is due to the fact that the volume of compressed air resulting from the compression of a unit volume of free air decreases with increase of the compression ratio (see above under "Volumetric Efficiency").

The volume of compressed air delivered at altitudes decreases at a slower rate than the barometric pressure. This relation is shown by the two ideal indicator cards in Fig. 3; one of a compressor working at sea-level, with an inital pressure P_1, the other at an altitude with initial pressure P_2.* The initial volume V and the final gage pressure P are the same for both compressors, P_3 and P_4 being the respective final absolute pressures. V_1 and V_2 are the final volumes, corresponding to the dotted isothermal curves, these volumes being taken as the basis, because they are those to which the compressed air will eventually shrink on losing the heat of compression. From Chapter II,

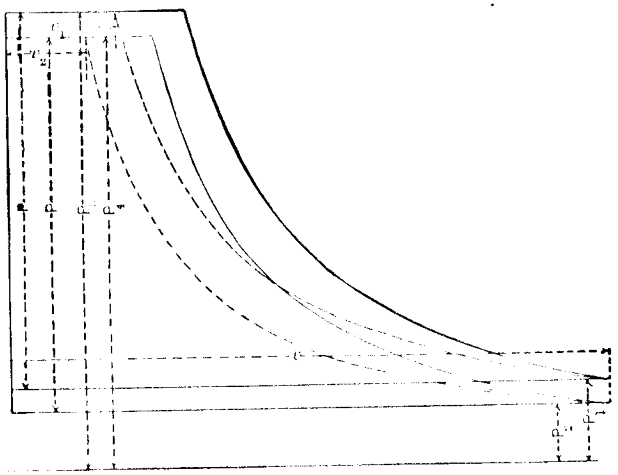

Fig. 3.

$$VP_1 = V_1P_3, \quad \text{or} \quad \frac{V}{V_1} = \frac{P_3}{P_1} \tag{1}$$

and

$$VP_2 = V_2P_4, \quad \text{or} \quad \frac{V}{V_2} = \frac{P_4}{P_2} \tag{2}$$

---

* The demonstration here given and the accompanying table are based upon an article by F. A. Halsey.

But since $P_3 = P_1 + P$, and $P_4 = P_2 + P$, equations (1) and (2) may be written:

$$\frac{V}{V_1} = \frac{P_1 + P}{P_1} = 1 + \frac{P}{P_1} \tag{3}$$

and

$$\frac{V}{V_2} = \frac{P_2 + P}{P_2} = 1 + \frac{P}{P_2} \tag{4}$$

Dividing equation (3) by equation (4):

$$\frac{V_2}{V_1} = \frac{1 + \frac{P}{P_1}}{1 + \frac{P}{P_2}}, \quad \text{or} \quad V_1 : V_2 :: 1 + \frac{P}{P_2} : 1 + \frac{P}{P_1} \tag{5}$$

Equation (5) gives the ratio between pressure and volume at sea-level and for any altitude at which the barometric pressure is P_2. Thus, if $P_2 = 10$ lb., $P = 90$ lb., and V_1 (from Table IV, Chapter X) = 0.1404 cu.ft., V_2 is found to be 0.0999, or about 0.1 cu.ft.

In Table I, columns 4 and 5, are given the relative volumetric outputs, at gage pressures of 70 and 90 lb., of a compressor working at different altitudes, the figures being percentages of the normal output at sea-level. These percentages are derived from equation (5), a constant loss of initial pressure of 0.75 lb. being assumed, to allow for the resistance of the inlet valves; that is, the sea-level pressure is taken as 14, instead of 14.7 lb. Although there is a difference of 20 lb. between the two assumed gage pressures, the outputs vary only by a few thousandths and may often be neglected.

Owing to the required increase of piston displacement per indicated horsepower (columns 8 and 9, Table I), it is customary to make compressor air cylinders for mountain work of larger diameter for the same size of steam cylinder than those for sea-level service. As against the losses in the air end of compressors at high altitudes, there is some gain in mean-effective steam pressure because the exhaust is against a lower atmospheric pressure. The same is true in part of the exhaust of machines using the compressed air. But the resultant of these gains is small and has little weight in offsetting the losses in the compressing cylinder, and those due to the lower calorific power of fuel at high altitudes.

The relation between compressor output and barometric pressure may be expressed in another way. Take two compressors of the same

TABLE I

Altitude, ft.	Barometric pressure		Relative output for gage pressure		M.E.P. for gage pressure		Cu.ft. piston displacement per I.Hp. for gage pressure		Cu.ft. compressed air per I.Hp. for gage pressure	
	Inches mercury	Lb. per sq.in.	70 lb.	90 lb.	70 lb.	90 lb.	70 lb.	90 lb.	70 lb.	90 lb.
1	2	3	4	5	6	7	8	9	10	11
0	30.00	14.75	1.000	1.000	33.1	38.2	6.93	5.99	1.111	.801
1,000	28.88	14.20	.967	.966	32.6	37.6	7.03	6.09	1.123	.787
2,000	27.80	13.67	.935	.933	31.1	36.9	7.15	6.20	1.103	.773
3,000	26.76	13.16	.904	.900	31.5	36.3	7.27	6.31	1.084	.759
4,000	25.76	12.67	.873	.869	31.0	35.6	7.39	6.43	1.065	.746
5,000	24.79	12.20	.843	.839	30.5	35.0	7.51	6.55	1.046	.733
6,000	23.86	11.73	.813	.809	30.0	34.3	7.65	6.67	1.028	.720
7,000	22.97	11.30	.785	.780	29.4	33.7	7.80	6.79	1.011	.708
8,000	22.11	10.87	.758	.751	28.9	33.1	7.94	6.92	.994	.695
9,000	21.29	10.46	.731	.723	28.3	32.5	8.09	7.06	.976	.683
10,000	20.49	10.07	.705	.696	27.8	31.8	8.24	7.20	.959	.670
11,000	19.72	9.7[illegible]	.680	.671	27.4	31.2	8.39	7.34	.942	.658
12,000	18.98	9.34	.656	.647	26.9	30.6	8.54	7.49	.925	.646
13,000	18.27	8.98	.632	.623	26.3	30.0	8.71	7.64	.908	.635
14,000	17.59	8.65	.608	.600	25.8	29.4	8.88	7.80	.891	.624
15,000	16.93	8.32	.585	.576	25.3	28.8	9.06	7.96	.875	.613
16,000	16.31	7.99	.562	.553	24.8	28.2	9.23	8.12	.859	.602

size, one operating under an atmospheric pressure of 14 lb. and the other at 10 lb. (corresponding approximately to an altitude of 10,000 ft.) If the first compressor works at 6 compressions, the final absolute pressure will be 14 × 6 = 84 lb. or about 70 lb. gage. To produce the same gage pressure the other compressor must work to an absolute pressure of 70 + 10 = 80 lb., the number of compressions being $\frac{80}{10} = 8$. Per cubic foot of free air, the first compressor will produce $\frac{1}{6}$ cu.ft. of compressed air; the second, $\frac{1}{8}$ cu.ft. Hence the ratio of the outputs is $\frac{1}{8} \div \frac{1}{6} = \frac{3}{4}$, or 0.750; while the ratio of the barometric pressures is $\frac{10}{14} = 0.714$.

Isothermal Horsepower Factors.—As stated in Chapter X, it is customary to use these factors for comparing the performance of compressors, especially when operating at altitudes above sea-level. The factors (Fig. 4) are based upon the barometric pressure and a theoretical air cylinder delivery of 100 per cent. They are derived from the equation:

Isothermal I.Hp. per 100 cu. ft. free air per min.

$$\frac{\text{Mean Effective Pressure} \times 100 \times 144}{33{,}000} = \frac{P \times \text{hyp. log. } R}{2.29},$$

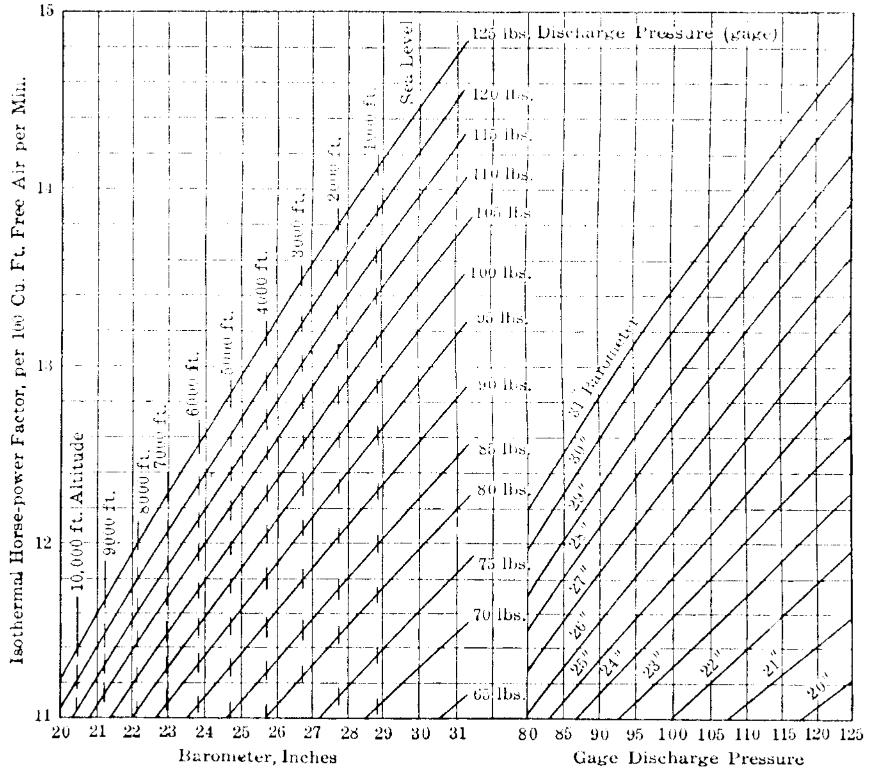

FIG. 4.—Isothermal Horsepower Factors Corresponding to Different Barometric and Gage Pressures (S. B. Redfield)

in which P = initial absolute pressure and R = absolute ratio of compression, that is, the absolute discharge pressure divided by P.

To find the Isothermal Compression Efficiency of a compressor working under a given barometric pressure, the factor as computed above, is divided by the actual I.Hp. of the steam cylinder, per 100 cu.ft. of free air compressed and delivered per minute. It must be remembered, in this connection, that a low horsepower per cubic foot of cylinder displacement may simply mean a small air output, due to low volumetric efficiency.

Stage Compression at Altitudes above Sea-level.—The greater the altitude the greater is the difference between delivery pressure and atmospheric pressure; that is, the ratio of compression is greater. For example, to produce a gage pressure of 75 lb. at 5,000 ft. elevation, 7.15 compressions are necessary. At sea-level this number of compressions would give a gage pressure of $(11.7 \times 7.15) - 11.7 = 90.4$ lb. Therefore, it is as reasonable to employ stage-compression for 75 lb., at 5,000 ft. elevation, as for 90 lb. at sea-level.

It is evident that the percentage volumetric loss from clearance increases with the altitude, because the piston must advance farther before the clearance air re-expands to a pressure below the diminished atmospheric pressure. In a stage compressor, there is practically but one clearance space, that in the intake cylinder. The value of the intercooler also increases with the altitude, because of the added heat due to the larger compression ratio.

For the above reasons, stage compression is, in general, advantageous for altitudes above sea-level; and the percentage of saving increases with the altitude.

Mechanically-controlled Air Valves were formerly used for some compressors. Mechanical control was best applied to inlet valves, to minimize the effect upon their action of inertia and spring resistance (see Chapters IX and XIII in previous editions of this book), and produced its best results at altitudes above sea-level. Since an ordinary inlet valve opens by difference of pressure (Chapter VII), the reduction in volumetric efficiency of the air cylinder due to valve resistance evidently becomes greater at altitudes, because the ratio of this resistance to the pressure of the intake air increases as the atmospheric pressure decreases.

Before the introduction of thin-plate valves mechanical control had some importance, but it is now obsolete.

CHAPTER XIII

EXPLOSIONS IN COMPRESSORS AND RECEIVERS

Explosions in air compressors and receivers occur with sufficient frequency to demand careful attention. Though they are attributable to ignition of volatile constituents of the lubricating oil, the immediate causes of this combustion are somewhat obscure. But, since explosions occur only in dry compressors, light may be thrown upon the subject by considering the conditions affecting the use of the lubricant. In Chapter IV attention was called to the fact that, if the cylinder temperature of a dry compressor rises too high, not only does proper lubrication become difficult, but the oil itself may be decomposed. Ignition unattended by actual explosion is probably frequent; the discharge pipe near the compressor sometimes becomes red-hot, and ignition has even extended into the receiver without producing a destructive explosion. The discharge-valve chests and passages, and the pipe leading from the compressor to receiver, often contain a black, sooty residue from decomposition of the lubricant. On passing with the compressed air into the receiver, the volatile constituents of the oil thus liberated would make a mixture of air and gas capable of producing an explosion. The extreme violence of such explosions is probably due in part to the high air pressure in the valve passages, discharge pipe, and receiver, since in high-pressure air combustion is more active than in air at atmospheric pressure.

Probable Causes.—As a number of compressor explosions have occurred at collieries, the effects of the presence of coal dust in the intake air of the compressor have been considered. Such a deposit in the valve passages, together with the sooty residue from decomposition of the oil, might produce a condition favorable to an explosion. A spark caused by the friction of the compressor piston, if working dry, or the continual passage of air at a high temperature over the carbonaceous deposit, might produce spontaneous combustion, and ignite the inflammable mixture of oil-vapor and air.* However, there are enough cases where explosions have occurred at mines and works other than collieries to prove that explosions are not necessarily dependent upon the presence of coal dust in the intake air. When the compressor is improperly situated in a room close to the boilers, some coal dust might be present in the air; but, though possibly assisting in the explosion, the quantity could hardly be large enough to produce by itself the observed results.

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* T. G. Lees, *Trans. Federated Inst. Mining Engineers*, Vol. XIV, p. 568.

The primary cause of compressor explosions is undoubtedly to be found in the working conditions prevailing in the cylinder. In single-stage dry compressors very high temperatures are often reached, due to poor design of the air cylinder, or running too fast (as when the compressor is too small for its work), or attempting to produce too high a pressure. The temperature of the discharge air from a single-stage compressor is found by the formula given in Chapter X:

$$T'' = T\left(\frac{P''}{P}\right)^{\frac{n-1}{n}=0.29}$$

in which T and P are the absolute initial temperature and pressure of the intake air; T'' and P'', the absolute final temperature and pressure; and n, the constant 1.41. Under normal conditions near sea-level, when the temperature of the atmosphere is 70° F., $P = 14$ lb., and the gage pressure at discharge, 80 lb., the final temperature would be:

$T'' = 70 + 459°\left(\frac{80 + 14}{14}\right)^{0.29} = 917°$ F. absolute, or 458° F. by the thermometer.

In using this formula, the compression is supposed to be purely adiabatic, no account being taken of loss of heat by radiation or of any cooling effect from the water-jackets. Little heat can in any case be abstracted by the jackets of a single-stage compressor. Air is a poor conductor, and the volume in the cylinder is not long enough under the influence of the jackets to be much affected by them. In compressors of this type the chief office of the jackets is to keep down the temperature of the cylinder walls and prevent the lubricating oil from being carbonized. It is probable that in a single-stage dry compressor, even if well designed and in good order, the discharge temperature generally ranges from 375° to 425° F., and may go higher.

In view of these considerations the quality of the lubricating oil used in the air cylinder, and especially its flashing- and ignition-points, are matters of importance.* The flashing-point of ordinary cylinder oil may be taken as from 330°–525° F. "An average of determinations on 40 samples of heavy oils having an average flash-point of 360° F., gave an average burning-point of 398° F. High flash-test cylinder oils, from 500°–560° F., gave burning-points of 600°–630° F."* Common lubricating oils flash at about 250° F., and kerosene, sometimes carelessly used for cleaning valves, at 150° F. or below. In the case of one explosion the flash-point of the cylinder oil was found to be only 295° F.† It would appear, therefore, that an explosion in a compressor cylinder, directly traceable to decomposition of the lubricant, is possible under normal conditions only when inferior, light mineral oils are used.

To produce an explosion there must be a sufficient increase of temperature to cause ignition of the lubricating oil or other combustible. In endeavoring to account for abnormal compressor temperatures, different theories have been advanced.

Some engineers have held that high cylinder temperatures may result from leakage of delivery valves, or past the piston; the argument being that the hot, high-pressure leakage air raises the initial temperature of the cylinderful of air to be compressed on the next stroke, so that the final temperature becomes abnormally high. It would appear that this reasoning does not take account of the fall in temperature due to re-expansion of the leakage and clearance air behind the piston on the intake stroke. As it may fairly be assumed that the compression cycle is approximately adiabatic, the fall in temperature of the re-expanded air (disregarding the heating effect of the hot cylinder surfaces) would nearly correspond to the original rise of temperature due to the compression of this air. This theory, therefore, does not seem tenable. Other causes may possibly exist, but we have no definite knowledge, in fact, as to what does take place in an air cylinder working hot enough to produce an explosion. In the absence of exact data, some light on the subject may be obtained from a study of the following examples.

Examples of Explosions.—An explosion which took place in one of the receivers of a compressor at the Clifton Colliery, England, attracted much attention, and is so instructive that some of the details are given here.‡ The air from the compressor passed to a series of three large receivers, the first being 7 ft. in diameter by 40 ft. long. While running apparently under normal conditions the safety valves of the receivers suddenly began blowing off with a deafening roar. Flames several feet high issued at great pressure from the safety valves, and sparks were blown out at the joints of the 8-in. pipe leading from the compressor to the first receiver. The air main near this receiver was nearly red-hot. That the receivers did not burst was thought to be due to the relief afforded by the four safety valves two on the first receiver and one on each of the others and to the fact that the underground engines driven by compressed air continued running for some minutes after the compressor was stopped. On examining the first receiver, after it had cooled, it was found that, just below the point where the air entered from the compressor, a mass of black carbonaceous matter had been deposited, from $1\frac{1}{2}$ to 2 in. thick and 6 sq.ft. in area. On analysis this showed the following composition, in per cent: volatile matter, 55.8, fixed carbon, 37.3, and ash, 6.9. The material was charred and had the appearance of hard vulcanite. A thin coating was noticed on the sides of the receiver (though only near the inlet pipe) and also in the pipe itself. The other two receivers were free from deposit. A carbonaceous coating, to a thickness of $\frac{1}{4}$ in. was found on the discharge valves and passages. The cylinder and piston surfaces were not dry and, though they showed signs of excessive heat, were uninjured.

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* The flashing-point of oil is the lowest temperature at which it gives off combustible vapors in sufficient quantity to be ignited by contact with flame. The ignition-point is the temperature to which the vapors must be raised in order to continue to burn.

* Alex. M. Gow, *Engineering News*, March 2d, 1905, p. 224.

† John Morison, *Trans. North of England Inst. Min. Engs.*, Vol. XXXVIII, p. 6.

‡ T. G. Lees, *Trans. Federated Inst. Mining Engineers*, Vol. XIV, pp. 555–559.

The gage pressure was usually 60 lb., which, with adiabatic compression, corresponds theoretically to a final temperature of 405° F., the temperature of the intake air from the engine-house being 80°. The lubricating oil used was guaranteed to have a flash-point of 554°, and ignition-point of 600° F. As the cylinders were water-jacketed, the discharge air should not, in regular working, reach these temperatures; in fact, readings previously taken from a thermometer in the outlet pipe showed that it usually registered about 350° F. It is significant, however, that on a previous occasion the mercury rose above 500°, but, as the thermometer tube burst, the temperature at the time of the explosion was not known. Afterward a pyrometer was fixed on the outlet pipe close to the discharge valves, and the temperature was found to range generally from 400° to 420° F., varying with the speed of the engine and the air pressure produced. Even with these temperatures, high as they are, it would seem impossible that ignition of the lubricating oil used could take place. It is evident that an unusual increase of temperature in the air cylinders must be accounted for, but no satisfactory explanation of this explosion has been offered.

A violent explosion occurred in the discharge pipe of a four-stage Laidlaw-Dunn-Gordon compressor, at a plant of the H. C. Frick Coke Co., Brownfield, Pa., which was furnishing air at 1,000 lb. pressure for air locomotives. The compressor was not damaged, though a large hole was blown in the pipe. It was thought that too much cylinder oil had been used, the record showing the consumption during the 5 months preceding the explosion to be 12 gal. per month. The average for the preceding year was 52.2 gal., but the reduction, great as it was, seemed to have been insufficient.*

Evidence as to another cause of trouble was obtained when a second explosion in the same compressor took place two years later. A recording thermometer, which had been installed in the discharge pipe close to the compressor, generally registered from 230° to 250° F., seldom exceeding 240°. A fusible plug, designed to blow out at between 325° and 350° F., was also set in the discharge pipe near the compressor. The monthly consumption of oil was further reduced to only 3.72 gal., a solution of castile soap and water being used almost exclusively for internal lubrication, with very good results.

Previous to the second explosion, the compressor had been running normally. The day before, the maximum temperature was 240° F., the thermometer generally registering between 190° and 230°. On the day of the explosion, the temperature reached 250° between 8 and 9 a.m. By 11 a.m. it was evident that something was wrong, the temperature almost reaching 270° at 11.15. Investigation showed that the fourth-stage discharge valves were out of order, but the engineer thought that by careful running he could finish the day. He held the temperature between 250° and 265° until 2.50 p.m., when the explosion occurred; the chart of the recording thermometer then showing 270°, followed by a high peak in the curve. Coincident with the explosion, the fusible plug melted and blew out, releasing the pressure and checking the temperature at 620°. The compressor, which was uninjured, was stopped, and a new plug put in, taking about 15 minutes, during which time the temperature dropped to 245°. On starting again (in doing which the engineer assumed an unnecessary risk) the temperature rose to 270°, before the compressor was shut down at 4.10 p.m. New valves and seats were put in, and on starting again two days later the temperature ranged from 220° to 240°.†

This explosion was ascribed by the management to "churning" of the air, due to leaky discharge valves, which allowed the high-pressure air to re-enter the fourth-stage cylinder. It is possible, however, that the valves of the cylinder were not working at all when the explosion took place, in which case the compressor became temporarily a three-stage machine. If this be true, in compressing to 1,000 lb. in three cylinders, the compressor was working under conditions for which it was not designed, and for which the cooling arrangements of the three remaining cylinders were inadequate. This explanation does not appear to be unreasonable.

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* It is probable that gummed oil and carbonaceous deposit had accumulated liberally wherever it could lodge in the interior of the compressor. R. P.

† Abstracted from a paper in *Mines and Minerals*, Vol. XXXII, p. 651, by William L. Affelder, Gen. Mgr. Bulger Block Coal Co., Bulger, Pa.

During the construction of the New York Aqueduct a fire occurred in a compressor receiver at one of the shafts. The air pressure was 80 to 90 lb., and the receiver, set outside of the engine-house, was exposed to the hot sun. Part of the discharge pipe leading to the receiver became red-hot. On stopping the compressor and cooling down the receiver, the entire inner surface of the latter was found to be coated with carbonaceous matter at least ¼ in. thick. Further investigation brought out the fact that the poppet discharge valves had sometimes occasioned trouble by sticking, and the engineer had been in the habit of using a squirt-can of kerosene to cut the gummy material clogging them. As the kerosene had a low flash-point, it was quickly vaporized, and when the cylinder temperature reached a sufficiently high point the explosion took place.

In a case at Butte, Mont., two duplex compressors, with air cylinders, respectively, of 32¼ by 60 in. and 24¼ by 48 in., and running at 50 rev. per min., were forcing air at 80 lb. pressure through a single 8-in. pipe. As somewhat over 1,200 cu.ft. of compressed air per minute were being produced, the velocity of flow would be nearly 3,500 ft. per min., or 58 ft. per sec. It had been noticed several times that a portion of the discharge pipe close to the compressor became red-hot. In the pipe between the compressors and receivers were several sharp bends, which increased the friction due to the rapid flow of the air. The receivers were always extremely hot. On one occasion the shaft timbering, 40 or 50 ft. below the shaft mouth, took fire from the hot air pipe.

Although the observed results of this explosion were localized in the discharge pipe, it is probable that oil was first vaporized either in the cylinder or when the compressed air was passing through the delivery valves; that a portion of it became hot enough to ignite, and in turn ignited an accumulation of oil vapor in the discharge pipe to the receiver.

It seems necessary to hold that the primary cause of explosion is to be looked for in the cylinder, not in the discharge pipe or receiver. That is, it is reasonable to assume that the conditions leading to explosion are initiated at the point of maximum pressure (and therefore of maximum temperature), which is toward the end of the stroke and while the air is passing through the discharge valves. If this temperature is high enough, vaporization of some of the lubricating oil will occur, followed by ignition, which might extend into the mixture of air and oil vapor in the discharge pipe.

Foul or poisonous gases may result from ignition of the lubricant in compressors or receivers, not necessarily followed by actual explosion. In an article in the *Transactions of the American Institute of Mining Engineers*, Vol. XXXIV, p. 158, an instance is noted of combustion in an air pipe and receiver. The compressed air was being used in an imperfectly ventilated upraise in a mine, 1,200 ft. from the compressor, and two men lost their lives, while four others barely escaped asphyxiation.

Other more or less similar cases are familiar to most miners, where foul air from the exhaust of machine drills has been observed; sometimes merely disagreeable, though often actively deleterious. The use of poor cylinder oil is probably responsible for this, as its lighter constituents may begin to volatilize and burn at a normal working temperature. Even if not actually fried on the hot metal surfaces, a low-grade oil will undergo a slow combustion or oxidation, which may produce enough carbon monoxide to raise materially the percentage of that poisonous gas in the confined atmosphere of working places of mines.

Mode of Using Lubricant for Air Cylinders of Compressors.

Sight-feed lubricators, as commonly employed for steam cylinders, are best. On the Clifton Colliery compressor, mentioned above, ordinary oil-cups were used, holding about ¼ pint; they were filled four times per day of 10 hours. With these oil-cups, if improperly adjusted, it would be possible for all the oil to be sucked into the cylinder within a few strokes after being filled. Such a result might be inferred, indeed, in this case, because of the large quantity of carbonaceous matter—oil, coal dust, etc.—found in and around the discharge valves and in the receiver. The oil feed should be carefully regulated, and a smaller quantity used in an air cylinder than a steam cylinder of the same size—say, one-third as much. Excess of oil increases the tendency to gum the valves. For stage compressors of ordinary size, 1 drop of good cylinder oil every 4 to 5 minutes is sufficient.

The periodical use of soap and water (soap-suds) is recommended for any compressor that can not be shut down at short intervals for overhauling. It is fed into the air cylinder through an oil-cup, say during one day per week. Or it may be forced in by an oil-pump, with which the compressor should be provided. Soap and water is a poor lubricant, and must be used more freely than oil, but it is effectual in cleansing the cylinder, valves, and ports from carbonaceous or gummy matter. If the compressor is to be stopped, as at the end of a shift, the feeding of soap and water should be discontinued some time before shutting down, and the oil-feed resumed, to avoid formation of rust. Every compressor should be overhauled from time to time, and thoroughly cleaned in all parts, especially around the valves and passages capable of furnishing a lodgment for oil or partly oxidized carbonaceous material.

Precautions for Preventing Explosions: (1) Always inclose the inlet valves in a cold-air box, connecting with the outside air, to avoid taking air from the hot engine-room. This conduces to economy in working, and by keeping down the final temperature tends to prevent decomposition of the oil. (2) The largest possible area of cylinder surface should be water-jacketed, including the cylinder heads. A liberal supply of the coldest water obtainable should be used for the jackets. The advantages in this respect of employing stage compression, with large inter- and aftercoolers, are undoubted. (3) Use only the best cylinder oil, with high flash- and ignition-points and in as small quantity as is consistent with proper lubrication. (4) Keep the valves clean. In the design of the compressor there should be no recesses or pockets, around valves or air passages, where oil could accumulate. (5) Never introduce kerosene into the cylinder for cleaning the valves while the compressor is running. (6) Arrange the air intake so that coal dust will not be drawn into the cylinder with the inlet air. (7) Place a thermometer in the discharge pipe, close to the cylinder, so that the engineer can watch the temperature, and stop or slow down the compressor if the temperature of the discharge air rises too high. A continuously-recording thermometer is to be recommended.

Conclusions.—Though compressor explosions are not uncommon, it is undoubtedly true that, while mixtures in certain proportions of air and oil vapor are explosive, oil is often burnt in the cylinders without causing a destructive explosion. This is proved by the frequent presence in air cylinders of sooty, carbonaceous deposits. The theories aiming to account for the observed phenomena by placing the responsibility entirely on leakage of discharge valves, or on "churning" of the air back and forth in the cylinder, due to sticking of the valves, are not conclusive nor satisfactory for the reasons already stated regarding the adiabatic compression cycle. The whole subject is at present obscure.

Before attempting to formulate conclusions, it would be desirable to secure more data. Definite knowledge of the conditions leading to explosions can be obtained by making laboratory investigations under controlled conditions, and applying them to the circumstances assumed to have caused explosions. In this way, it could be determined to what extent, if any, the cylinder temperature is raised by leaky discharge valves, or by "churning" of the air. The effect of small particles of relatively non-conducting material, as carbon (coal dust) or lint, drawn into the cylinder with the intake air, could also be investigated. Such carbonaceous points might become incandescent, due to heating from small jets of flame from burning oil, and remain so long enough to ignite larger volumes of mixed air and oil vapor. It may be suggested that research leading to fuller knowledge of this subject might be carried out in the mechanical engineering laboratories of a university, or at the works of a compressor builder.

CHAPTER XIV

AIR COMPRESSION BY THE DIRECT ACTION OF FALLING WATER

Principle.—When air in small bubbles is intimately mixed with water, the water breaks into foam, through which the bubbles tend to rise and escape. But if the mixed air and water are drawn downward by a strong falling current, as in a vertical pipe, the air is compressed. Then if, after reaching the depth and head of water-column required to produce the compression desired, the direction of flow is changed to the horizontal and the velocity diminished, the bubbles of compressed air are liberated and may be collected in a suitable chamber. The air pressure in this chamber corresponds to the effective head of water, that is, the depth of the water level in the chamber below the level in the outflow or tail-race. Thus, in this method of air compression, no piston, valves, nor other moving parts, are necessary.

The combined specific gravity of the mixture of air and water in the compressing pipe is less than that of the water in the return column. Therefore, the head required to overcome friction and to produce flow must be greater than if the apparatus were merely an inverted siphon, and as the difference in weight increases with depth (and air pressure produced) the motive head, or difference in level between the surfaces of water at inlet and in tail-race, must be correspondingly increased.

As the bubbles of air are minute and thoroughly disseminated through the water during its descent, the total cooling surface is very large and isothermal compression results. The moisture-carrying capacity of the air is therefore smaller than if the air were compressed adiabatically. During compression the percentage of moisture in each globule of air first increases until the point of saturation is reached; then, on further compression, the excess moisture is deposited, so that when re-expanded the air is relatively dry. This method was first tested on a working scale in 1877 and 1878, by J. P. Frizell, of Boston, in a 36-ft. shaft at the Falls of St. Anthony, on the Mississippi River, near St. Paul.

Design.—A complete demonstration of the problem is complicated and the numerical values of some of the factors are not yet accurately

known. For practical purposes, it is sufficient to consider the behavior of the mixed water and air in the downward moving or induction column and in the return column, from which most of the entrained air has been liberated; these two columns are the principal elements of the plant.

Induction Column.—In this, the volume of entrained air depends upon the volume of water and the velocity of flow. Some of the air during compression is dissolved in the water, which at saturation contains about $\frac{1}{20}$ of its volume of air, independent of pressure.

In the following analysis:

Let p = atmospheric pressure, pounds per square inch;
p' = pressure of the compressed air, pounds per square inch;
Q = available compressed air, in terms of cubic feet of free air;
R = ratio of total volume of entrained air to volume of water;
S = weight of one cu.ft. free air at pressure p, pounds;
d = diameter of induction pipe, feet;
V = assumed mean velocity of downward flow in induction pipe, feet per second;
m = mean weight of mixed air and water in induction pipe, pounds per cubic foot;
c = volume of entering water, cubic feet per second;
c_1 = mean volume of mixed air and water passing down the induction pipe, cubic feet per second.

As the dissolved air in the descending column is compressed, its volume diminishes; hence, to maintain saturation, more of the entrained air must be taken up by the water, until at the bottom of the column the original 0.05 cu.ft. of free air in solution becomes $0.05\frac{p}{p'}$ cu.ft. at pressure p'; the consequent drain on the entrained air being then $0.05\left[\frac{p'}{p} - 1\right]$ cu.ft. The total volume of air required to be entrained at the top of the column is therefore $c[R - 0.05]$ cu.ft. per sec.

On reaching the bottom of the column,

$$Q = c\left[R - 0.05\frac{p'}{p}\right], \quad \text{whence} \quad R = \frac{Q}{c} + 0.05\frac{p'}{p}$$

also, it may be deduced that

$$m = \frac{62.5 + S(R - 0.05)}{2}\left[\frac{1}{1 + R} + \frac{1}{1 + R\frac{p}{p'}}\right],$$

$$c_1 = \frac{c}{2}\left[1.9 + R\left(1 + \frac{p}{p'}\right)\right],$$

and

$$d = \sqrt{\frac{c_1}{0.7854V}} = 1.128\sqrt{\frac{c_1}{V}}.$$

Assuming that air bubbles of about ¼-in. diameter will rise in still water 0.8 ft. per sec., the rising velocity of bubbles of compressed air, on reaching the top of the induction column, would be $0.8\left(\frac{p'}{p}\right)^{\frac{1}{6}}$ ft. per sec. (the velocity varying as the square root of the bubbles' diameter). Hence, to carry down the bubbles, the actual mean velocity of downward flow of the induction column must be

$$V_1 = V + 0.8\left(\frac{p'}{p}\right)^{\frac{1}{6}}.$$

The losses of head in the induction column are: (1) the head required to entrain air at the top, estimated as 1 ft. $= h$; (2) velocity head, taken as $0.0155V_1^2 = h_1$; (3) entrance head, covering the *vena contracta*, is half the velocity head, or $0.0078V_1^2 = h_2$; (4) friction head, in the entire length H of the column, is $\frac{H}{2}\left(\frac{4V^2 + 5V - 2}{14,400}\right) = h_3$; (5) head lost by the dispersion of the downward flowing column in the separating chamber at the foot is $\frac{mc_1}{2g}V_1^2$. The net work done by the induction column therefore

$$= mc_1\left[H - (h + h_1 + h_2 + h_3) - \frac{V_1^2}{2g}\right] \text{ ft. lb.}$$

Return column.—Let a be the area of the ascending column, square feet and c_2 the mean volume of the column, cubic feet per second; whence, the mean velocity $V_2 = \frac{c_2}{a}$. If the column be of circular section, with diam. d_2, $V_2 = 1.273\left(\frac{c_2}{d_2^2}\right)$. The mean volume c_2 of upward flow

$$= c_1\left[1 + 0.025\left(\frac{p'}{p} - 1\right)\right] \text{ cu. ft. per sec.}$$

The mean weight of the column is

$$m_1 = \frac{62.5 + 0.05\left(\frac{p'}{p} - 1\right)}{2}\left[1 + \frac{1}{1 + 0.05\left(\frac{p'}{p} - 1\right)}\right] \text{ lb. per cu.ft.}$$

The losses of head due to resistance in the return column and the work done are: (1) the velocity head $h_4 = 0.0155(V_2 - 0.8)^2$; (2) friction head $h_5 = \frac{l}{d_2}\left(\frac{4V_2^2 + 5V_2 - 2}{7,200}\right)$, where l is the height of the column in feet and 7,200 is a constant assumed for the rough walls of an unlined shaft (for smooth walls, the constant would be 14,400, as already used for the induction column); (3) pressure head $h_6 = 2.31\ (p' - p)$; (4) work of isothermal compression of Q cu.ft. free air is $W = 144\ pQ$ hyp. log. $\frac{p'}{p}$ (see Eq. 4, Chapter II).

Hence, the total resistance and work done in the return column $= m_1c_2(h_4 + h_5 + h_6) + W$ ft.-lb. As this must be equal to the total useful work of the induction column,

$$mc_1\left[H - (h + h_1 + h_2 + h_3) - \frac{V_1^2}{2g}\right] = m_1c_2(h_4 + h_5 + h_6) + W.$$

The efficiency is the ratio of the useful work to the work of the falling water, or $E = \frac{W}{62.5c(H - h_6)}$.

The last two equations contain all the essential factors entering into the design of an installation for hydraulic compression.*

Construction Details.—These are illustrated typically by the pioneer plant for the hydraulic compression of air, which was erected in 1896 by the Taylor Hydraulic Air Compressing Co., of Montreal, for the Dominion Cotton Mills, Magog, Province of Quebec.† In a 128-ft. shaft (Fig. 1) was erected a vertical compressing pipe a, 3 ft. 8½ in. in diameter, the lower part increasing to 4 ft. 8 in., and made of $\frac{5}{16}$-in. steel plate. This pipe passes through the bottom of a receiving cham-

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* For further details, see papers by J. P. Frizell, *Journal of Franklin Institute*, Vol. 140, p. 145, and A. E. Chodzko, *Mining and Scientific Press*, Dec. 16, 1916, pp. 879–883.

† The description given here is based on an article in the *Canadian Engineer*, March, 1897, and information furnished to the author by the builders. See also *Engineering and Mining Journal*, Dec. 26th, 1896, p. 606, and *Railway and Engineering Review*, Sept. 17th, 1898, p. 513.

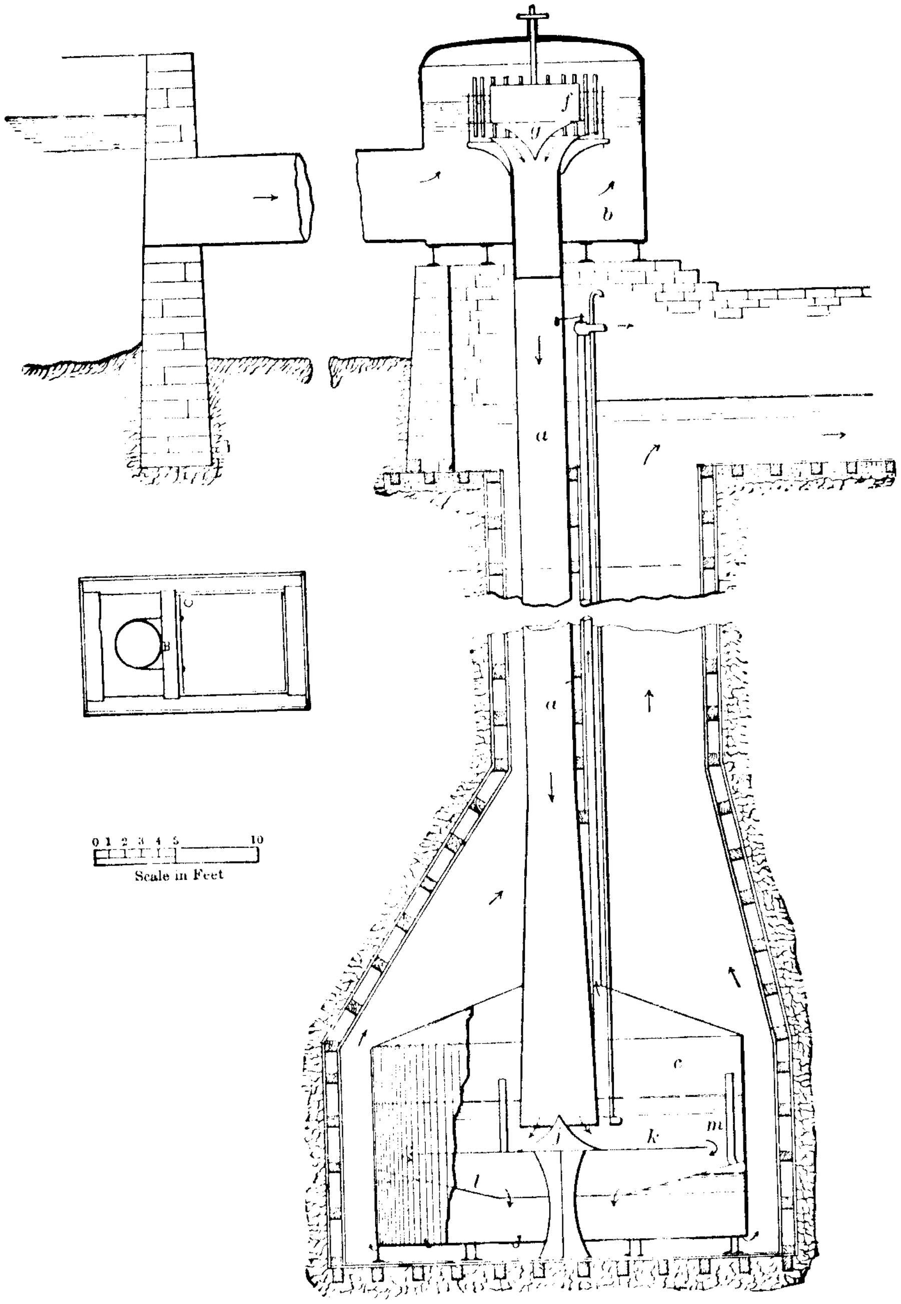

FIG. 1.—Taylor Hydraulic Air Compressor, Magog Plant.

ber *b*, 12 ft. in diameter by 12 ft. high, to which water is conducted from a head-race. Water flows into and fills the pipe, which extends nearly to the bottom of the shaft. Through a series of small feed pipes, air is drawn with the water into the top of the main pipe and is compressed while being carried down the shaft. The compressed air col-

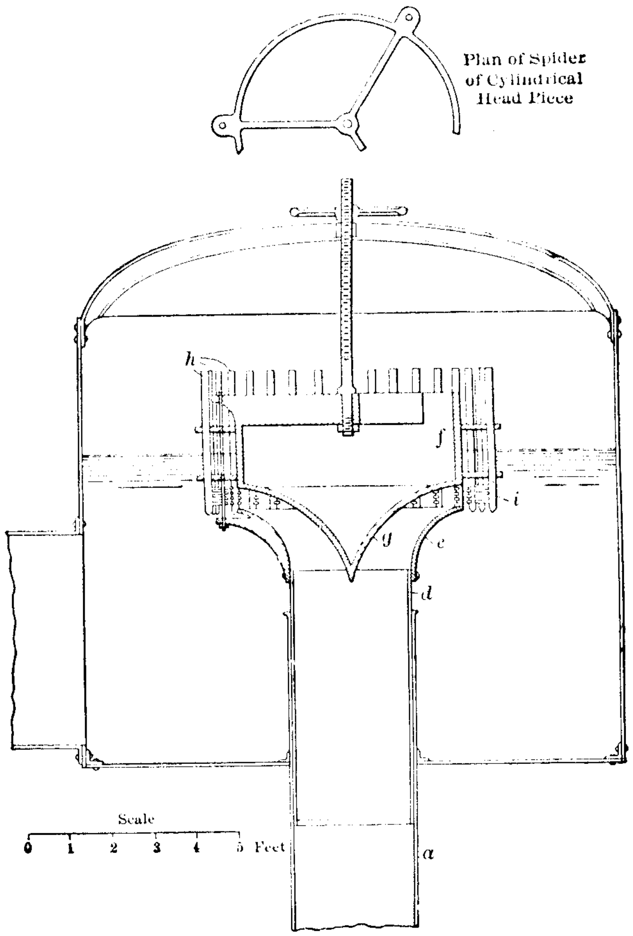

FIG. 2.—Head Piece, Magog Plant.

lects in a chamber *c*, while the water is returned to a tail-race near the top. The difference of level between intake and tail-race is about 22 ft., which produces the requisite speed of flow. Into the top of the vertical pipe *a* is inserted a telescoping section of pipe *d* (Fig. 2), carrying a bell-mouth *e* and headpiece *f*, terminating below in an inverted conoid *g*. Between *e* and *g* is an annular opening, through which the water enters the compressing pipe. The headpiece carries thirty 2-in. pipes *h*, *h*, 4 ft. long, open at the top and closed at the bottom. Into each of these pipes are screwed 32 short horizontal $\frac{3}{8}$-in. pipes *i*, *i*, all directed into the annular opening between *e* and *g*. As the entering water passes among the small pipes air is entrained, carried down the main pipe in the form of bubbles, and is thus compressed.

Near the bottom of the shaft the compressing pipe enters the "separating" chamber *c*, 17 ft. in diameter and 12 ft. high, open below and supported upon legs which raise it 16 in. above the shaft bottom. Within this chamber is a conoidal "disperser" *jk*, 12 ft. in diameter. Below is an apron *l*, 5 ft. wide. When the water, charged with air bubbles, reaches the disperser it is first directed outward, then deflected by the apron toward the center, and finally escapes through the open bottom of the separating tank into the return column. During this process of travel the compressed air separates from the water, most of it collecting in the upper part of chamber *c*. Part of the air collects in the annular space under the apron, and joins the main body of air through the pipe *m*.

Tests on air compressed by water to 135 lb. pressure have shown it to contain only 17.7 per cent oxygen, or 3 per cent less than that in atmospheric air. This is ascribed to absorption of oxygen by the water during compression. The shortage of oxygen does not affect the breathing of the miners, though candles burn with difficulty; the physiological effect of the gases from blasting is more serious.

TABLE I.—TESTS ON THE MAGOG PLANT*

No. of Test	Water discharged, cu.ft. per min.	Available head, ft.	Available hp.	Air delivered, cu.ft. per min. at atmosphere pressure	Air pressure, lb. per sq.in.	Actual hp. of compressor	Efficiency, per cent
1	6122	21.4	247.7	1377	52	132.5	53.5
2	5504	21.9	228.0	1363	52	131.0	57.5
3	4005	22.3	168.9	1095	52	105.3	62.4
4	7662	21.1	305.9	1616	52	155.4	50.8
5	6312	21.7	260.0	1506	52	144.8	55.7
6	7494	21.2	299.8	1560	52	150.2	50.1

Temperatures during tests: external air, 70–80°; water, 75.2–80°; compressed air, 75.2–80°.

* Tests made by Prof. C. H. McLeod, of McGill University, Aug., 1896; published in *Engineering and Mining Journal*, Dec. 26, 1896, p. 606. See also *Engineering and Mining Journal*, July 9, 1910, p. 59; Sept. 10, 1910, p. 568.

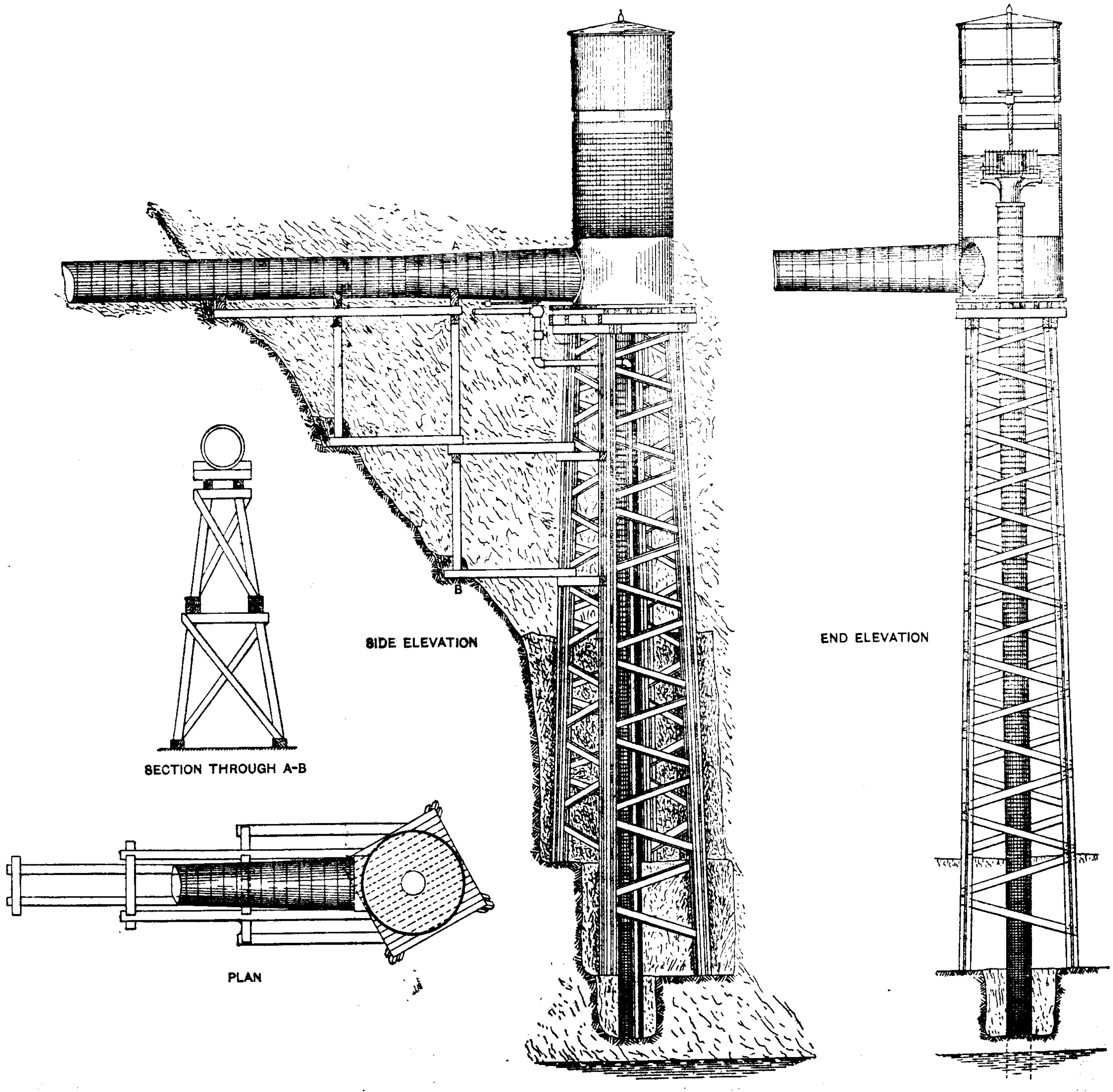

Fig. 3.—Hydraulic Air-compressing Plant at Kootenay.

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The parts were incorrectly proportioned in this first installation, and the efficiency could be increased by using a larger air chamber, to prevent air from going to waste.

Kootenay Plant. In 1898 1900 a Taylor plant was built for the Kootenay Air Supply Co., Ainsworth, B. C. The topographical conditions are such that a high head is obtained without sinking a deep shaft. From a dam the water is carried in a wooden-stave pipe, 5 ft. in diameter and 1,354 ft. long, over a short trestle built against the side of a gorge, to the receiving tank. The latter, 17 ft. in diameter by 20 ft. high, is placed on a wooden tower, 110 ft. high (Fig. 3). From the tank the pressure pipe, 33 in. in diameter, descends to the ground level and then down a shaft 105 ft. deep. After compressing the air the water returns up the shaft to the tail-race at the creek level. Fig. 4 shows the details of the receiving chamber at the bottom of the shaft.

The effective compressing head is 107 ft., the total height of the intake pipe being over 200 ft. This produces a high velocity of flow and large delivery of compressed air. The compressed air main, 9 in. in diameter, is 2 miles long, carrying from 4,200 to 4,600 cu.ft. of free air per minute. Branch service pipes convey the air to neighboring mines, where it is used for rock-drills and other machinery. On the basis of 600 hp., represented by the volume and pressure of the air, the cost of the entire plant, including pipe lines, was about $100 per horsepower.

Victoria Plant was completed in 1906 at the Victoria Copper Mine, Rockland, Ontonagon County, Mich. The local conditions led to a novel mode of installation. The water is conducted from a dam on the Ontonagon River through a 4,700-ft. canal, furnishing a head at the terminal forebay of 72 ft. above the river-level. Three independent units are built side by side at 19-ft. centers in a vertical shaft 340 ft. deep. In the original design, the subdivision of the air, as admitted at the intake head (Fig. 5) was carried farther than in either of the plants described above, by inserting 1,800 $\frac{3}{8}$-in. horizontal feed pipes, in the series of larger vertical pipes encircling the inverted cone.

After the plant was put in operation, serious trouble was experienced by the freezing up of the small pipes of the intake heads, due to the severe winter climate of the region. This led to the removal of the heads, the water being allowed simply to flow into the top of the compression pipes. The breaking up and agitation of the mass of water, in changing its direction of flow from the forebay into the compressing pipes, entrained the air quite efficiently, and it is stated that the capacity of the plant, in cubic feet of free air compressed per minute, is practically the same as when the intake heads were in use.

The compressing pipes are 5 ft. in diameter, lined with concrete;

and separating cones and dispersers, also of iron and concrete, are built in a chamber at the bottom. In this chamber, 281 ft. long and 18 ft. by 21 ft. average cross-section, the compressed air is trapped and thence

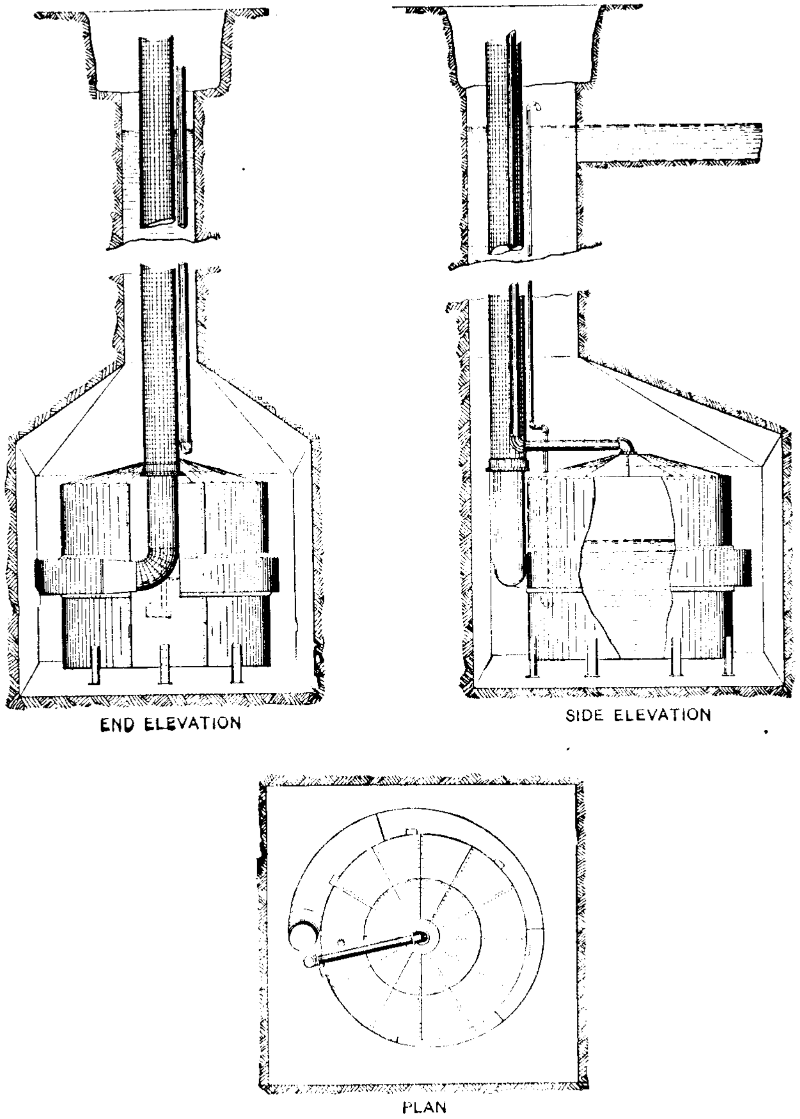
END ELEVATION
SIDE ELEVATION
PLAN

Fig. 4.—Hydraulic Air Compressor at Kootenay.

drawn off through a 24-in. main. The compressing water, flowing down the intake pipes, stands normally at a level about $14\frac{1}{2}$ ft. below the roof of the chamber, thus leaving an air capacity of about 80,000

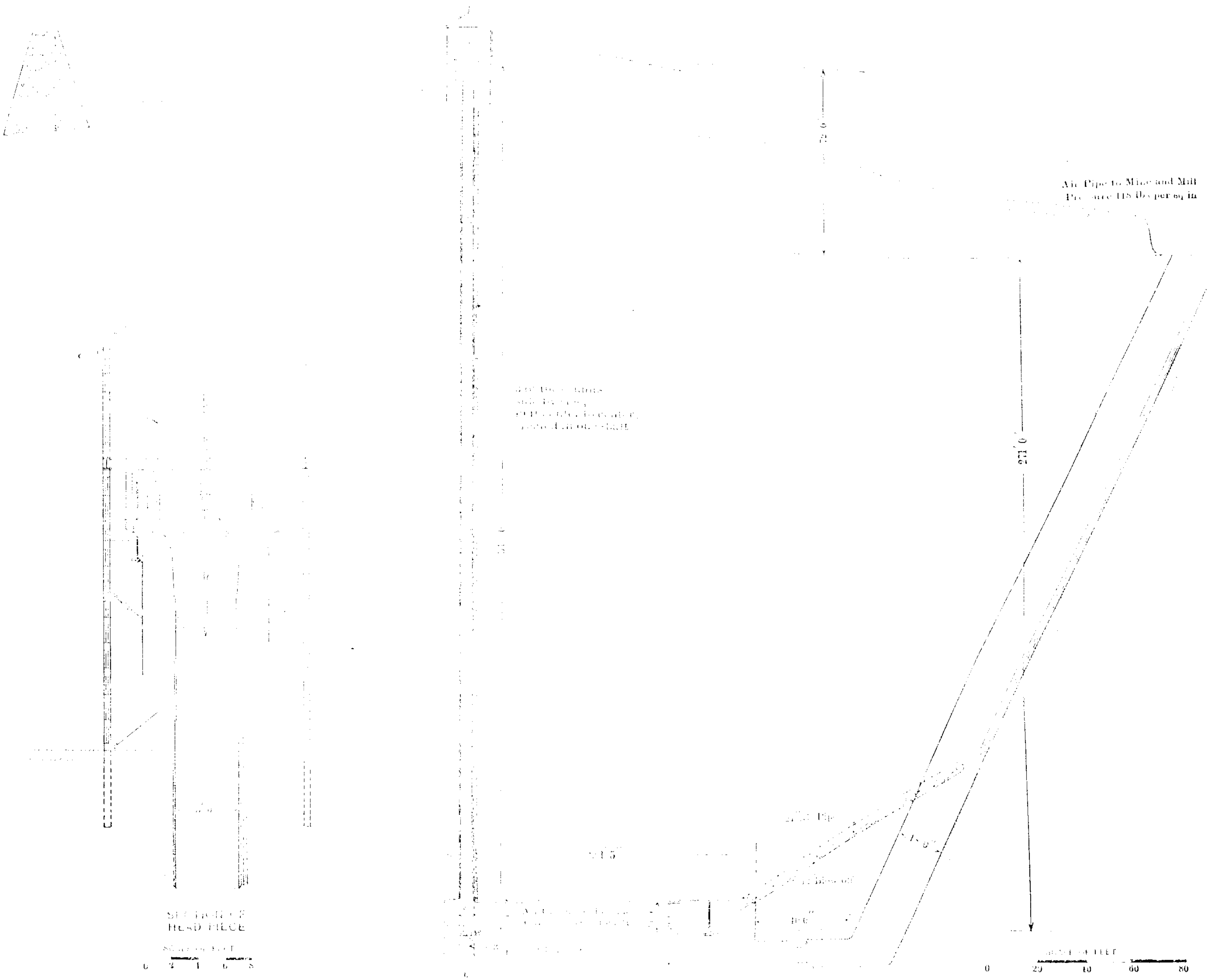

Fig. 5.—Hydraulic Air-compressing Plant. Victoria Mine, Mich.

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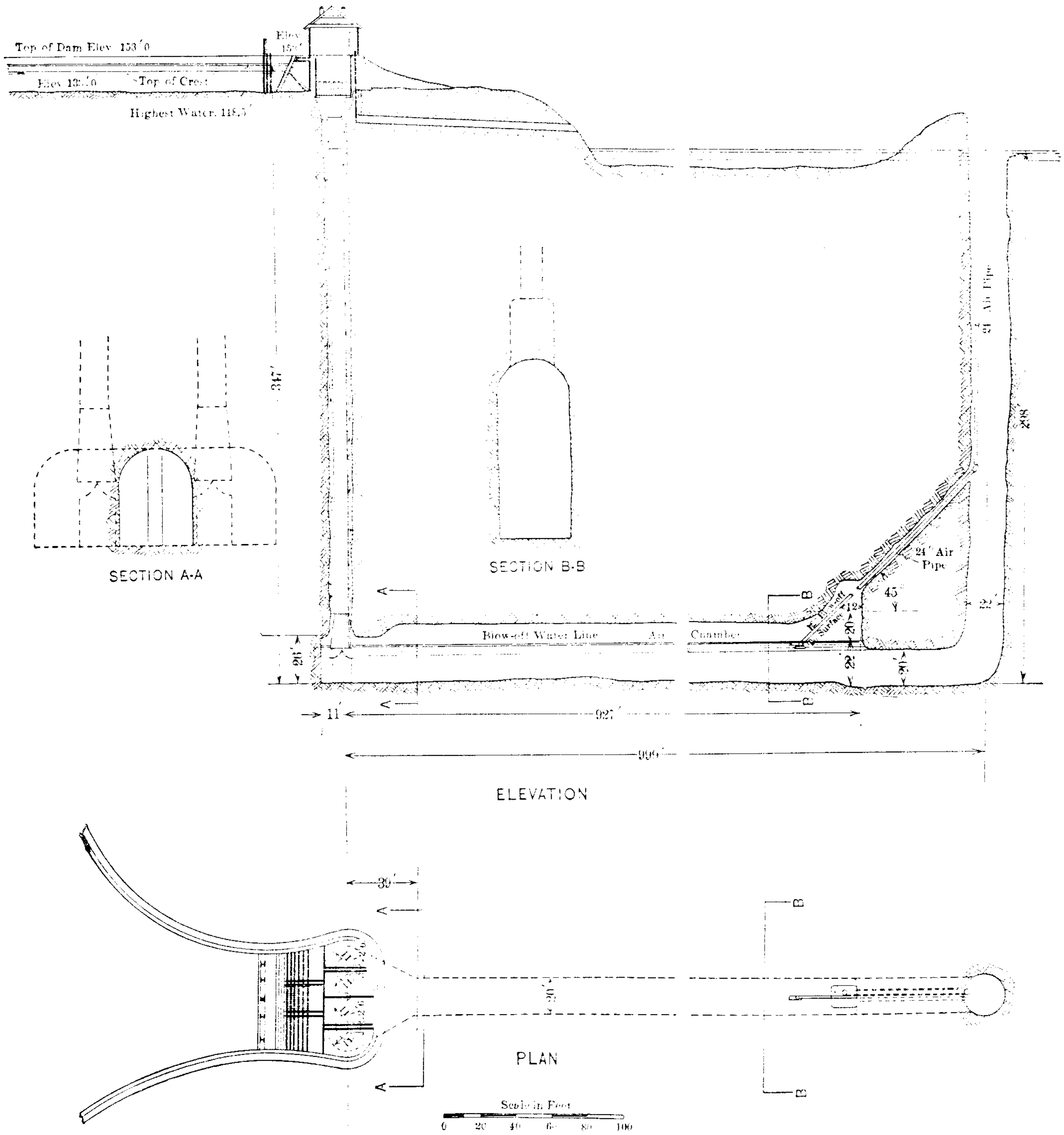

Fig. 6.—Cobalt Power Co's Plant, Plan and Elevation.

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cu.ft. Connected with the end of the air chamber is an inclined shaft, 270 ft. in vertical depth, through which the water returns to the surface. The tail-race from this shaft is 72 ft. below the level of the intake, this height measuring the motive head producing the flow of water. Thus the air in the underground chamber is under a pressure due to 270 ft. head of water, or 118 lb. per sq. in.

For regulating the operation of the original plant a pipe passed from the air chamber up the compressing shaft to the surface, whence branches were led to the intake heads. The compressed air conveyed in this regulating pipe operated a device connected with each intake head, whereby the latter was automatically raised above the water-level in the receiving tanks whenever the air pressure exceeded the normal, thus stopping the flow of air through the feed pipes. A 12-in. blow-off pipe passes from the water-level in the air chamber to the mouth of the inclined shaft carrying the return water column. If air to the full compressor capacity is drawn off, the water-level in the air chamber rises as the air pressure falls, thus sealing the lower end of the blow-off pipe; then, when the consumption of air decreases, the pressure in the chamber rises, depressing the water-level until the blow-off orifice is uncovered, when more air is blown off. Thus the working pressure is maintained within quite narrow limits. The great size of the air chamber—corresponding to the receiver of an ordinary air-compressor—gives a large storage capacity.

Table II.—Air Measurements, Victoria Plant

Area, sq.ft.	Velocity, ft. per sec.	Cu.ft. per min.	Absolute pressures		Hp.
			Free air, lb.	Compressed air, lb.	
4	44.09	10,580	14	128	1,430
4	49.74	11,930	14	128	1,623
4	38.50	9,238	14	128	1,248

Water Measurements

Flume area, sq. ft.	Velocity, ft. per sec.	Cu.ft. per min.	Head, ft.	Hp.	Efficiency, per cent
71.75	3.033	13,057	70.5	1,741	82.17
67.03	3.684	14,820	70.0	1,961	82.27
72.16	2.936	12,710	70.6	1,700	73.50

When all three compressing units are in operation, with a total capacity of from 34,000 to 36,000 cu.ft. of free air per minute, about 70,000 cu.ft. of air per minute may be drawn off for a period of 18 minutes, without causing a drop in pressure of more than 5 lb. For each unit, the output ranges from 9,000 to 12,000 cu.ft. per minute, and the volume of water used from 12,700 to 14,800 cu.ft. Tests made on a single intake head in May, 1906, by Prof. F. W. Sperr, gave the results shown in Table 11.

The air is used at the Victoria Mine for general power purposes at the mine and mill, including a 500-hp. hoisting-engine, and seven pumps. The cost per horsepower is about $2.25 per year, including all operating expenses. Over 1,000 hp. can be developed by the three compressing units.*

Cobalt Power Co.'s Plant. Following a series of efficiency tests made in 1909 on a large number of steam compressors at the silver mines in the Cobalt, Ont., district (see latter part of Chapter X), a Taylor compressor was built at Ragged Chutes, 9 miles from Cobalt, on the Montreal River. In a distance of 1,000 ft. there is a drop of 54 ft. From the forebay the water flows into two 16-ft. heads (Fig. 6), in each of which sixteen 14-in. vertical intake pipes are set in a horizontal disk. Below the disk the heads taper to 8 ft. 4¾ in., below which point they extend 15 ft., telescoping into the tops of 8½-ft. diameter concreted compression shafts, 330 ft. deep. To regulate the inflow, and adjust the position of the heads to the forebay water-level, the heads are suspended from two vertical hydraulic cylinders. The heads, with their large-diameter intake pipes, were designed to prevent freezing in the severe winter climate of the region (see above, under Victoria Plant).

The water with the entrained air flows through the heads at a velocity of 15 to 19 ft. per second. Due to the compression of the air in the shaft, this velocity gradually diminishes, with a further reduction in the lower 40 ft. of shaft, which is flared to 12½-ft. diameter. At the bottom of each shaft is a steel-plate-capped concrete diverting cone (see also Fig. 5 and accompanying description).

The shafts terminate in a tunnel (air chamber), 26 ft. high, 20 ft. wide and 1,000 ft. long, in which the air is completely liberated, and which serves as a receiver. So great a length was not required for these purposes, but was adopted to utilize the total head (54 ft.) of the stream. From the tunnel, air is drawn off through a 24-in. pipe passing in an inclined riser to the vertical tail shaft. The 12-in. blowoff pipe

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* For further details see article by D. E. Woodbridge, *Engineering and Mining Journal*, Jan. 19, 1907, p. 125

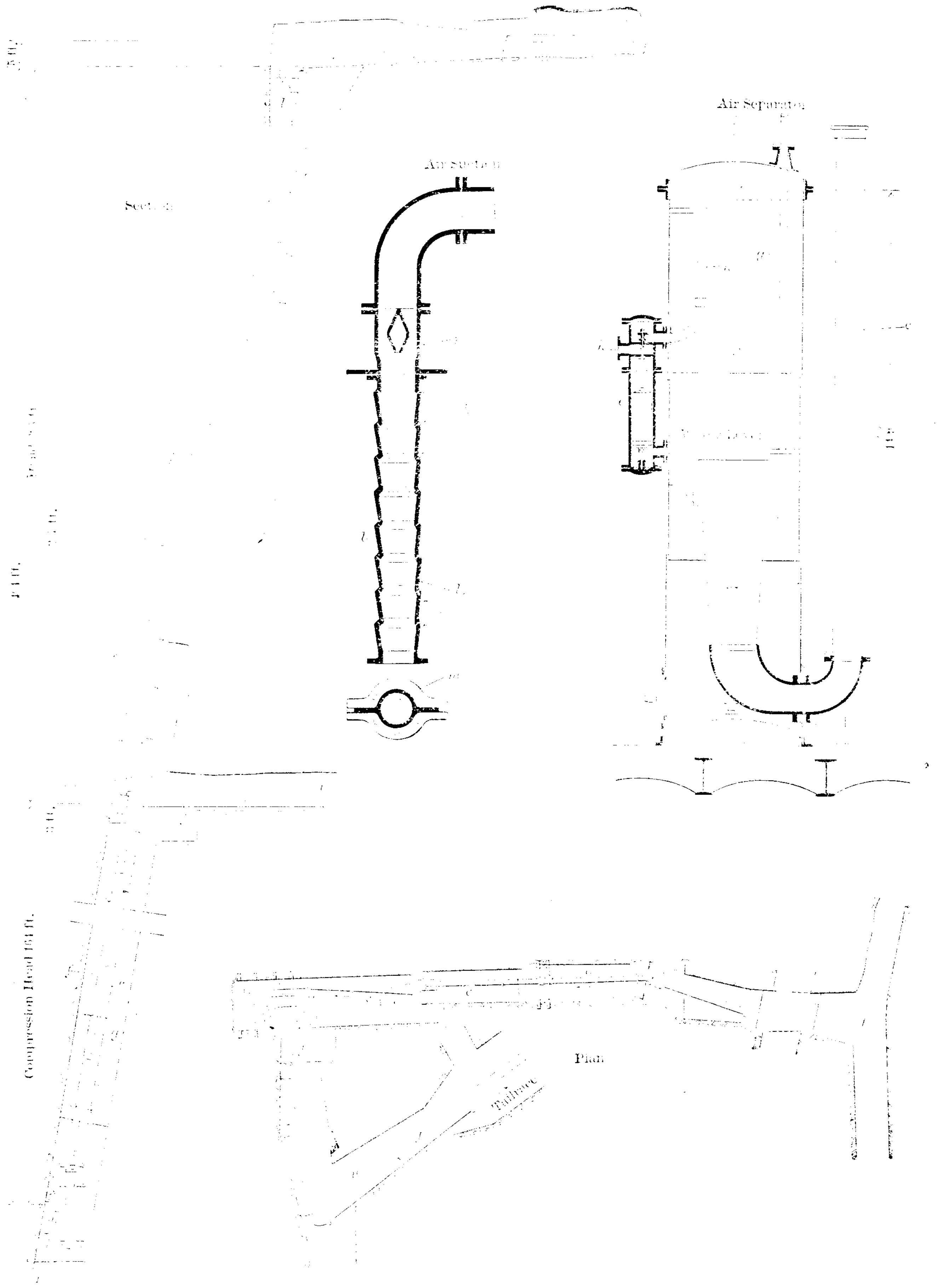

Fig. 7.—Underground Hydraulic Air-compressing Plant at Clausthal, Germany.

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acts in case the air pressure in the tunnel should force the water-level below the roof of the outlet to the tail shaft, and so cause fluctuations of pressure. The air pressure produced is that due to the net head of water in the tail shaft; in this case, 276 ft., corresponding to 120 lb. per sq.in.

This plant compresses 10,000 cu.ft. of free air per minute, corresponding to about 5,500 hp. The air is carried through 9 miles of 20-in. pipe to Cobalt. From there branch pipes connect with the different mines, the total piping (20, 12, 6 and 3-in.) being about 21 miles. Total cost, excluding piping, about $1,000,000, or $185 per horsepower. The air is sold by the company at 25 cents per 1,000 cu.ft. at 100 lb. pressure.*

Other Plants. – In the state of Washington there is a 200-hp. plant. Head of water, 45 ft., height of compressing pipe, 260 ft., diameter, 3 ft.; volume of water, 53 cu.ft. per second; air pressure, 85 lb.†

A small plant at Peterborough, Ont., has an 18-in. compressing pipe, in a 12-in. shaft. Depth of separating chamber below discharge level, 64 ft., air pressure, 25 lb.

Near Norwich, Conn., on the Shetucket River, there is a large plant for general power purposes.‡

In 1907–8 a hydraulic air compressor was installed at a silver mine at Clausthal, Germany. Fig. 7 shows the general design, with details of the intake head and compressing chamber. A flow of water in the tunnel *t* is led through an $8\frac{1}{2}$-in. cast-iron pipe *a*, to the air intake *b*, which consists of a number of flaring rings *l*, in the upper rim of each of which is a series of small holes *k*, for admitting the air. Additional inlet area is provided at the top of the intake by a nest of small curved pipes *m*. The mixed air and water pass into the $8\frac{1}{2}$-in. compression pipe *c*, 492 ft. long, laid in an inclined shaft, and discharging into the compressing chamber *d*. This chamber is 52 in. by 14 ft. 9 in. high. From a point near its top the compressed air passes through pipe *n* to the automatic check-valve *e*, and thence, by pipe *h*, to the receiver *i*. The water leaves the compressing chamber by the $8\frac{1}{2}$-in. pipe *f*, which discharges at a point 161 ft. above, into a tail-race occupying the mine level *u*. An equalizing discharge pipe *g*, from the compressing chamber, is led up the shaft, parallel to *f*, entering the latter at the level of the tail-race. Total cost of the plant is stated to be $3,750.§

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* C. H. Taylor, *Mines and Mining*, April, 1910, p. 532.

† *Engineering and Mining Journal*, April 27, 1901.

‡ *Compressed Air Magazine*, April, 1906, p. 3,980.

§ Abstracted from a description by P. Bernstein, in *Glückauf*, Mar. 14, 1908. Translation by E. K. Judd in *Engineering and Mining Journal*, Aug. 1, 1908, p. 228. See also *Zeitschr. Ver. Deutscher Ing.*, Nov. 5, 1910.

The average flow of water is 792 gal. per min.; which, falling through a vertical height of 325 ft., produces theoretically 66.3 hp. A flow of 845 gal. per min. gave 353 cu.ft. of air, at 71.2 lb. gage. To compress 1 cu.ft. of air adiabatically to this pressure requires 0.147 hp. and to compress 353 cu.ft., about 51.9 hp. Since 70.5 theoretical horsepower are produced by the flow of 845 gal. per min., the efficiency is $\frac{51.9}{70.5} = 73.6$ per cent.

The **first cost** of hydraulic air compressors is not excessive, compared with the cost of ordinary compressors of the same capacity, but in general the system is not suitable for small plants. Maintenance and running expenses are very low; no skilled attendance is required, and depreciation is nominal in substantially erected plants. By comparing the figures given in Tables I and II, it will be seen that at the Victoria Plant there was a marked increase in efficiency over previous installations, due to the greater motive head, and a more complete separation of the air from the water in the receiving chambers.

It has been suggested that it might be feasible to employ the system in connection with an ordinary compressor plant; that is, to produce a low air pressure by the water plant, and then to admit this air to the compressor cylinder where it would be brought to the required higher tension. In effect, this would be stage compression, in which the air would be cooled to normal temperature before entering the high-pressure cylinder.

Part Second

TRANSMISSION AND USE OF COMPRESSED AIR

CHAPTER XV

CONVEYANCE OF COMPRESSED AIR IN PIPES

THE diameter of the pipe is of vital importance, and when proportioned properly to the volume of air, and to the distance, the transmission losses are very small compared with the other losses incident upon air compression. Transmission losses appear in two ways: as direct loss of power, and as loss of pressure or head.

Loss of Power.—The warm air passing from the receiver into the air main soon assumes the temperature of the surrounding atmosphere; therefore, the calculation for transmission loss in long lines may be made without regard to the effect of temperature upon the volume of the air. The reduction in pressure by frictional loss in pipe transmission is accompanied by a proportionate increase in volume. Although the air pressure at the motor is diminished, there is no loss in the final volume of free air. As shown below, while the loss of pressure due to friction is small in piping of proper diameter, the actual loss of power is still smaller, because the pipe itself acts in a measure like a receiver—as a reservoir of power. In practice, much of the transmission power loss is due to leakage from joints and flaws in the pipe.

Loss of Pressure for short distances takes place according to the laws governing the flow of all fluids, varying directly as the length of pipe, directly as the square of the velocity, and inversely as the pipe diameter. For long distances the application of these laws becomes somewhat complex. Besides the factors just mentioned, it is necessary to take into account the volume and pressure of the air, and the initial and final pressures at the ends of the pipe line. In general, for a given diameter of pipe, when the volume of air discharged and its initial pressure remain constant, the loss of pressure is proportionate to the length of the pipe.

In actual service, however, the initial pressure and volume of discharge do not remain constant, and, in the passage of the air through the pipe, other modifying factors must be taken into account. In flowing through a long line of piping the pressure is gradually reduced by friction, while the volume is correspondingly increased. Therefore, to maintain in the pipe the flow of a given quantity of air, the volume of which is constantly increasing, the velocity also must increase, and this requires an increase of head or pressure.

The formulas commonly used assume that the loss of head is proportional to the length of pipe, and if a certain head is required to maintain the flow of a given quantity of air in a pipe 1000 ft. long, twice this head would suffice for a pipe 2000 ft. long. But, when the air has passed through the first thousand feet of pipe, part of its motive head has been lost; and as the volume has thereby increased, a greater head will be necessary to maintain the flow in the second thousand feet. The ordinary formulas do not take into account the increase of volume due to this loss of head. To transmit a given volume of air at a uniform velocity and loss of pressure would require a pipe of gradually increasing area. This, of course, is impracticable, and if the discharge is to be kept constant in pipe of uniform section, both volume and velocity must increase as the pressure is reduced by friction. The loss of head in properly proportioned pipes is so small, however, that in practice the increase in volume is usually neglected.

The discharge capacity of piping is not proportional to the cross-sectional area alone. Although the periphery is directly proportional to the diameter, the interior surface resistance is greater in a small than in a large pipe, because the ratio of perimeter to area is greater. To pass a given volume of air, a 1-in. pipe of given length requires over three times as much head as a 2-in. pipe of the same length. The character of the pipe also, and the condition of its inner surface, have much to do with the frictional resistance. The irregularities incident upon coupling together the lengths of pipe also increase friction. As the influences by which the values of some of these factors may be modified are not fully understood, the results obtained from formulas are only approximately correct.

D'Arcy's Formula, as adapted to compressed-air transmission, is:

$$D = c\sqrt{\frac{d^5(p_1 - p_2)}{w_1 l}}, \text{ or } D = \frac{c\sqrt{d^5}}{\sqrt{l}} \times \sqrt{\frac{p_1 - p_2}{w_1}}$$

in which D = the volume of compressed air, cubic feet per minute, discharged at the final pressure;

c = a coefficient varying with the pipe diameter, as determined by experiment;

d = nominal diameter of pipe, inches;*

l = length of pipe, feet;

p_1 and p_2 = initial and final gage pressures, pounds per square inch;

w_1 = density of the air, or its weight in pounds per cubic foot, at pressure p_1.

The formula in its factored form (see above) is convenient for use. Table I gives the values of c, d^5, and $c\sqrt{d^5}$; Table II, the value of w_1 for initial gage pressures up to 100 lb. per sq. in.; Table III, values of $\sqrt{\frac{p_1 - p_2}{w_1}}$ for terminal pressures of 20–100 lb., and pressure losses of

TABLE I

Diameter of pipe, inches	Values of c	Fifth powers of d	Values of $c\sqrt{d^5}$	Diameter of pipe, inches	Value of c	Fifth powers of d	Values of $c\sqrt{d^5}$
1	45.3	1	45.3	7	60.3	16,807	7,817
2	52.6	32	297	8	60.7	32,768	10,988
3	56.5	243	876	9	61.0	59,049	14,812
4	58.0	1,024	1,856	10	61.2	100,000	19,480
5	59.0	3,125	3,298	11	61.4	161,051	24,800
6	59.8	7,776	5,273	12	61.6	248,832	30,925

TABLE II

Gage pressure, pounds	w_1	$\sqrt{w_1}$	Gage pressure, pounds	w_1	$\sqrt{w_1}$
0	0.0761	0.276	55	0.3607	0.600
5	0.1020	0.319	60	0.3866	0.622
10	0.1278	0.358	65	0.4125	0.642
15	0.1537	0.392	70	0.4383	0.662
20	0.1796	0.424	75	0.4642	0.681
25	0.2055	0.453	80	0.4901	0.700
30	0.2313	0.481	85	0.5160	0.718
35	0.2572	0.507	90	0.5418	0.736
40	0.2831	0.532	95	0.5677	0.753
45	0.3090	0.556	100	0.5936	0.770
50	0.3348	0.578			

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* The actual diameters of wrought-iron pipe are not the same for all sizes as the nominal diameters. This difference is small except in the 1¼-in. and 1½-in. sizes, the actual diameters of which are 1.38 in. and 1.61 in. respectively.

1 to 10 lb.* Intermediate values are obtained by interpolation. No allowance is made for pipe leakage, nor for incidental friction due to bends in the pipe (Table VII).

Example.—Given a 5-in. pipe, 2,500 ft. long; how many cubic feet of air per min. at 70 lb. initial pressure can be transmitted, with a loss of pressure of 3 lb.?

From Table I, $c\sqrt{d^5} = 3{,}298$; from Table III, $\sqrt{\frac{p_1 - p_2}{w_1}} = 2.615$ and $\sqrt{l} = 50$. Substituting in the formula:

$$D = \frac{3{,}298}{50} \times 2.615 = 172.5 \text{ cu.ft. compressed air per min.}$$

Volumes of compressed air may be converted into free air by multiplying by the absolute pressure in atmospheres (1 atmos. = 14.7 lb.). Thus, 100 cu.ft. of air at 80 lb. gage, or 94.7 absolute pressure, correspond to 644 cu.ft. of free air, at sea-level. Table I, Chapter XII, gives the air pressures in pounds per square inch for altitudes to 15,000 ft., with the corresponding barometric readings.

Graphic Solution of D'Arcy's Formula. (Fig. 1.)† —The following problems may arise:

1. To find the diameter of pipe; given the volume of compressed air, length of pipe, initial pressure and maximum drop in pressure.

With a straight-edge, join the scales marked length of pipe and cubic feet of compressed air, and note the intersection on axis *A*. Join the initial pressure with the drop in pressure, and note intersection on axis *B*. A line joining these two points of intersection will cut scale No. 3 at the required pipe diameter.

2. To find the volume of compressed air that a pipe will carry; given the length and diameter of pipe, initial pressure, and maximum allowable drop in pressure.

Join the initial pressure with the drop, and note the intersection on axis *B*. Join this intersection with the pipe diameter, and note the intersection on axis *A* of a prolongation of this line. A line joining this point on *A* with the pipe length will cut scale No. 2 at the required volume.

3. To find the maximum length of pipe that will carry a given volume of air; given the pipe diameter, initial pressure, and maximum drop.

Join the initial pressure with the drop, and note intersection on axis *B*. Join this point with the pipe diameter, and note intersection on axis *A*. A line joining the point on *A* with the given cubic feet of air will cut scale No. 1 at the required length of pipe.

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* Reproduced by permission from *Compressed Air*, Feb., 1898, pp. 374–376.

† See paper by C. W. Crispell, *Trans.* Am. Inst. Min. Eng., Vol. LVIII, p. 93.

TABLE III (WILLIAM COX)

Values of $\sqrt{\frac{p_1 - p_2}{w_1}}$

Final pressure p_2, lb.	Losses of Pressure, $p_1 - p_2$									
	1 lb.	2 lb.	3 lb.	4 lb.	5 lb.	6 lb.	7 lb.	8 lb.	9 lb.	10 lb.
20	2.325	3.241	3.918	4.466	4.930	5.336	5.693	6.014	6.309	6.574
21	2.293	3.198	3.868	4.410	4.870	5.272	5.627	5.946	6.237	6.502
22	2.262	3.157	3.819	4.356	4.812	5.211	5.564	5.878	6.168	6.432
23	2.233	3.117	3.772	4.304	4.756	5.152	5.501	5.814	6.102	6.362
24	2.205	3.079	3.727	4.254	4.702	5.093	5.440	5.752	6.036	6.296
25	2.178	3.042	3.684	4.206	4.649	5.036	5.381	5.688	5.975	6.233
26	2.152	3.007	3.642	4.158	4.597	4.981	5.323	5.630	5.913	6.173
27	2.127	2.973	3.601	4.112	4.548	4.928	5.268	5.572	5.856	6.113
28	2.103	2.939	3.561	4.068	4.499	4.877	5.215	5.518	5.799	6.056
29	2.079	2.907	3.523	4.024	4.452	4.828	5.164	5.466	5.745	5.999
30	2.056	2.876	3.485	3.982	4.408	4.781	5.114	5.414	5.691	5.942
31	2.034	2.844	3.448	3.942	4.365	4.735	5.066	5.364	5.637	5.888
32	2.012	2.815	3.414	3.904	4.323	4.690	5.019	5.312	5.586	5.834
33	1.991	2.786	3.381	3.866	4.282	4.646	4.971	5.264	5.535	5.782
34	1.971	2.759	3.348	3.830	4.242	4.603	4.926	5.216	5.487	5.733
35	1.952	2.733	3.317	3.794	4.202	4.561	4.881	5.170	5.439	5.686
36	1.933	2.707	3.286	3.758	4.164	4.520	4.839	5.126	5.394	5.639
37	1.915	2.682	3.255	3.724	4.126	4.480	4.797	5.084	5.349	5.594
38	1.897	2.656	3.225	3.690	4.090	4.441	4.757	5.042	5.307	5.550
39	1.879	2.632	3.196	3.658	4.054	4.404	4.717	5.002	5.265	5.509
40	1.862	2.608	3.168	3.626	4.020	4.368	4.680	4.962	5.226	5.468
41	1.845	2.585	3.140	3.596	3.987	4.333	4.643	4.924	5.187	5.426
42	1.829	2.563	3.114	3.566	3.956	4.299	4.609	4.888	5.148	5.385
43	1.813	2.542	3.088	3.538	3.924	4.267	4.575	4.852	5.109	5.344
44	1.798	2.521	3.064	3.510	3.895	4.235	4.540	4.814	5.070	5.306
45	1.783	2.501	3.040	3.484	3.866	4.203	4.506	4.778	5.034	5.268
46	1.769	2.481	3.017	3.458	3.837	4.171	4.471	4.744	4.998	5.230
47	1.755	2.462	2.995	3.432	3.808	4.139	4.439	4.710	4.962	5.192
48	1.742	2.444	2.972	3.406	3.779	4.109	4.408	4.676	4.926	5.155
49	1.729	2.426	2.950	3.380	3.752	4.080	4.376	4.642	4.890	5.120
50	1.716	2.407	2.927	3.356	3.725	4.051	4.344	4.608	4.857	5.085
51	1.703	2.389	2.906	3.332	3.698	4.022	4.313	4.578	4.824	5.050
52	1.690	2.372	2.886	3.308	3.671	3.993	4.283	4.546	4.791	5.015
53	1.678	2.355	2.865	3.284	3.645	3.965	4.254	4.516	4.758	4.983
54	1.666	2.338	2.844	3.260	3.620	3.938	4.225	4.484	4.728	4.952
55	1.654	2.321	2.823	3.238	3.596	3.911	4.196	4.456	4.698	4.920
56	1.642	2.304	2.804	3.216	3.571	3.885	4.169	4.428	4.668	4.889
57	1.630	2.289	2.785	3.194	3.547	3.860	4.143	4.400	4.638	4.860
58	1.619	2.273	2.766	3.172	3.524	3.835	4.117	4.372	4.611	4.832
59	1.608	2.258	2.747	3.152	3.502	3.811	4.091	4.346	4.584	4.805
60	1.597	2.242	2.730	3.132	3.479	3.787	4.066	4.320	4.557	4.775

TABLE III—*Continued*

Values of $\sqrt{\frac{p_1 - p_2}{p_1}}$

Final pressure p_2, lb.	Losses of Pressure, $p_1 - p_2$									
	1 lb.	2 lb.	3 lb.	4 lb.	5 lb.	6 lb.	7 lb.	8 lb.	9 lb.	10 lb.
61	1.586	2.228	2.712	3.112	3.458	3.764	4.042	4.291	4.530	4.747
62	1.576	2.214	2.695	3.092	3.437	3.742	4.019	4.268	4.503	4.718
63	1.566	2.200	2.678	3.074	3.417	3.720	3.995	4.244	4.476	4.693
64	1.556	2.186	2.662	3.056	3.397	3.698	3.971	4.220	4.452	4.668
65	1.546	2.173	2.647	3.038	3.376	3.676	3.948	4.196	4.428	4.642
66	1.537	2.160	2.631	3.020	3.356	3.654	3.926	4.172	4.404	4.617
67	1.528	2.147	2.615	3.002	3.337	3.634	3.905	4.150	4.380	4.592
68	1.519	2.134	2.600	2.984	3.318	3.615	3.884	4.128	4.356	4.566
69	1.510	2.122	2.584	2.968	3.300	3.596	3.863	4.104	4.332	4.541
70	1.501	2.110	2.570	2.952	3.283	3.576	3.842	4.082	4.308	4.516
71	1.492	2.098	2.556	2.936	3.265	3.556	3.820	4.060	4.284	4.494
72	1.484	2.086	2.543	2.920	3.247	3.537	3.799	4.038	4.263	4.471
73	1.476	2.075	2.529	2.904	3.229	3.517	3.778	4.018	4.242	4.449
74	1.468	2.064	2.515	2.888	3.211	3.498	3.759	3.998	4.221	4.427
75	1.460	2.052	2.501	2.872	3.193	3.480	3.741	3.978	4.200	4.405
76	1.452	2.041	2.487	2.856	3.177	3.463	3.723	3.958	4.179	4.383
77	1.444	2.030	2.473	2.842	3.162	3.446	3.704	3.938	4.158	4.361
78	1.436	2.019	2.461	2.828	3.146	3.429	3.686	3.918	4.137	4.339
79	1.428	2.009	2.449	2.814	3.130	3.412	3.667	3.898	4.116	4.317
80	1.421	1.999	2.437	2.800	3.115	3.395	3.648	3.878	4.095	4.294
81	1.414	1.989	2.425	2.786	3.099	3.377	3.630	3.858	4.074	4.272
82	1.407	1.979	2.413	2.772	3.084	3.360	3.611	3.840	4.053	4.253
83	1.400	1.969	2.401	2.758	3.068	3.343	3.593	3.820	4.035	4.234
84	1.393	1.959	2.388	2.744	3.052	3.326	3.575	3.802	4.017	4.215
85	1.386	1.949	2.376	2.730	3.037	3.310	3.559	3.786	3.999	4.196
86	1.379	1.939	2.364	2.716	3.022	3.294	3.543	3.768	3.981	4.177
87	1.372	1.929	2.352	2.702	3.008	3.279	3.527	3.752	3.963	4.158
88	1.365	1.920	2.340	2.690	2.994	3.265	3.511	3.734	3.945	4.139
89	1.358	1.910	2.330	2.678	2.981	3.250	3.495	3.718	3.927	4.120
90	1.351	1.901	2.319	2.666	2.967	3.235	3.479	3.700	3.909	4.101
91	1.345	1.893	2.309	2.654	2.954	3.221	3.463	3.684	3.891	4.082
92	1.339	1.884	2.298	2.642	2.940	3.206	3.447	3.666	3.873	4.064
93	1.333	1.876	2.288	2.630	2.927	3.191	3.432	3.650	3.855	4.048
94	1.327	1.867	2.278	2.618	2.914	3.177	3.416	3.634	3.840	4.032
95	1.321	1.859	2.267	2.606	2.900	3.162	3.401	3.618	3.825	4.016
96	1.315	1.850	2.257	2.594	2.887	3.148	3.387	3.604	3.810	4.000
97	1.309	1.842	2.246	2.582	2.875	3.135	3.373	3.590	3.795	3.984
98	1.303	1.833	2.236	2.570	2.862	3.123	3.360	3.576	3.780	3.969
99	1.297	1.825	2.226	2.560	2.851	3.110	3.347	3.562	3.765	3.953
100	1.291	1.817	2.217	2.550	2.840	3.098	3.334	3.548	3.750	3.937

---

4. To find the pressure at which the air must enter the pipe; given, maximum allowable drop, volume of air, and the diameter and length of pipe.

Join the length of pipe with the volume of air, and note intersection on axis *A*. Join this point with the pipe diameter, and note intersection on axis *B*. A line joining the point on *B* with the allowable drop will cut scale No. 4 at the required initial pressure.

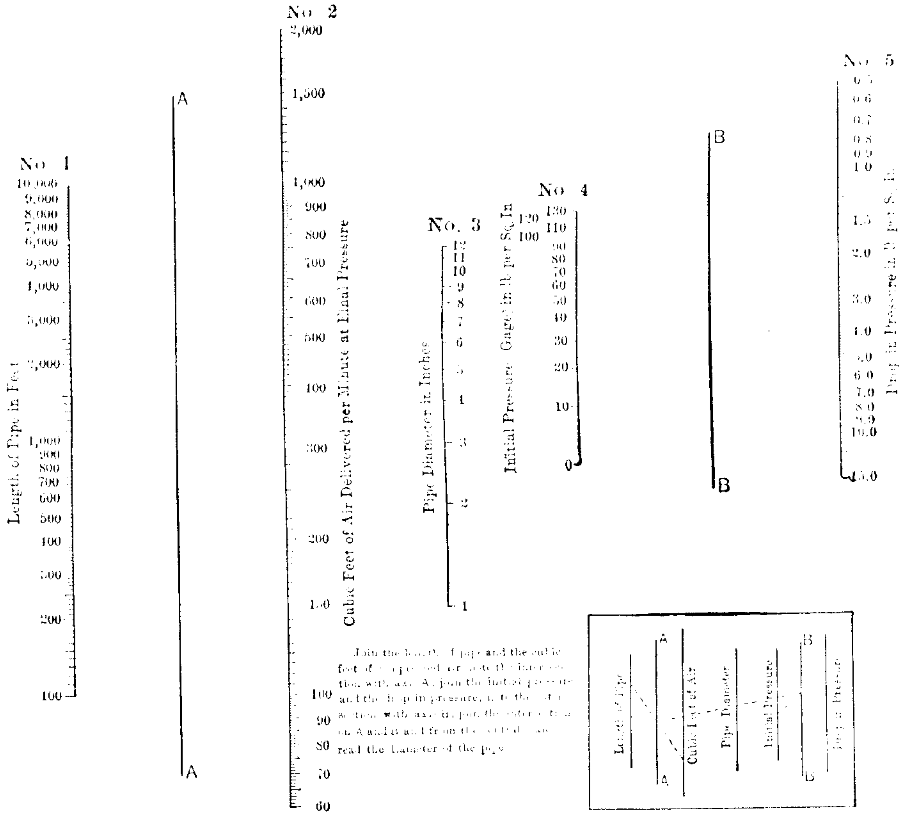

FIG. 1.—Nomogram for Finding Air Transmission Losses (Crispell).

5. To find the pressure drop; given, initial pressure, volume of air, and length and diameter of pipe.

Join the pipe length with the volume of air, and note intersection on axis *A*. Join this point with the pipe diameter, and note intersection on axis *B*. A line joining the point on *B* with the initial pressure will cut scale No. 5 at the required pressure drop.

In solving six problems by the nomogram, the mean error was less than 0.5 per cent.

NOTE.—The nomogram in Fig. 1 uses volumes of *compressed air*, whereas compressor capacity and the air consumption of rock-drills and other machines are usually expressed in terms of cubic feet of *free air*. Volumes of free air at different altitudes can be converted into volumes of compressed air at different absolute pressures by means of Fig. 2, which is based on the equation $PV = P'V'$ (Chap. II). The error in using this nomogram is from about −0.15 to +0.5 per cent.

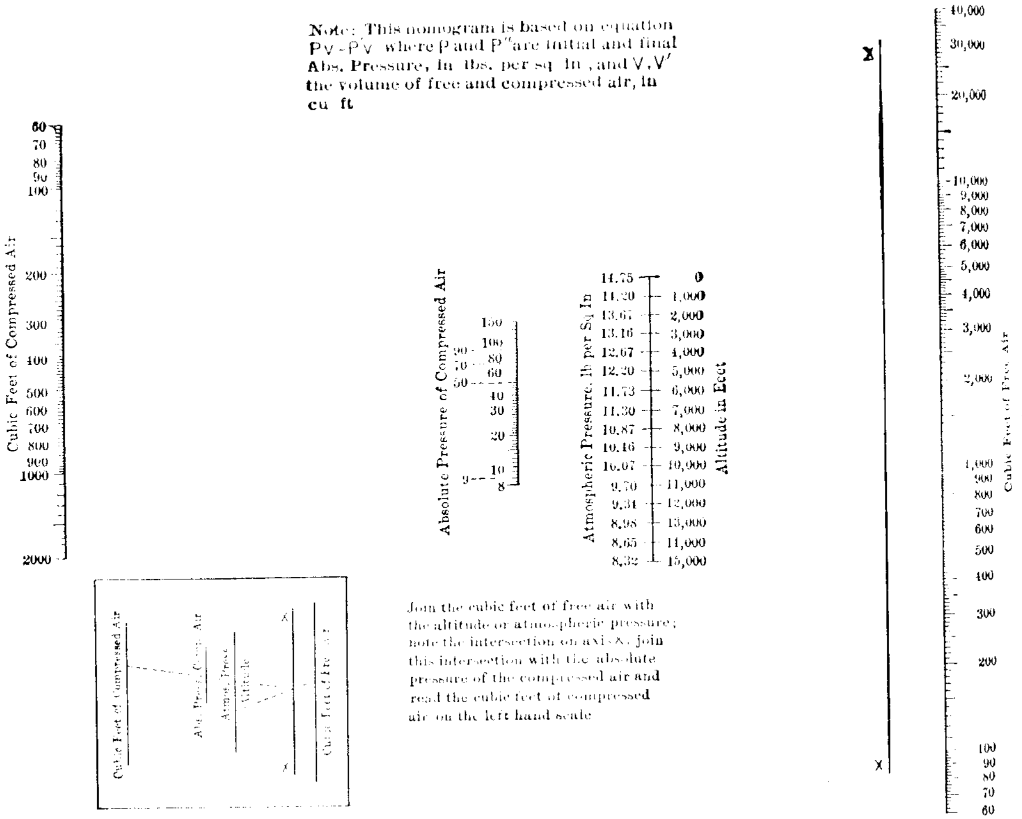
Note: This nomogram is based on equation Pv-P'v' where P and P'' are initial and final Abs. Pressure, in lbs. per sq. in., and V,V' the volume of free and compressed air, in cu ft
Cubic Feet of Compressed Air
60
70
80
90
100
200
300
400
500
600
700
800
900
1000
2000
Absolute Pressure of Compressed Air
150
100
90
80
70
60
50
40
30
20
10
9
8
Atmospheric Pressure, lb per Sq In
14.75
14.20
13.67
13.16
12.67
12.20
11.73
11.30
10.87
10.46
10.07
9.70
9.34
8.98
8.65
8.32
Altitude in Feet
0
1,000
2,000
3,000
4,000
5,000
6,000
7,000
8,000
9,000
10,000
11,000
12,000
13,000
14,000
15,000
X
X
Cubic Feet of Free Air
40,000
30,000
20,000
10,000
9,000
8,000
7,000
6,000
5,000
4,000
3,000
2,000
1,000
900
800
700
600
500
400
300
200
100
90
80
70
60
Cubic Feet of Compressed Air
Abs. Press. Comp. Air
Atmos. Press.
Altitude
X
X
Cubic Feet of Free Air
Join the cubic feet of free air with the altitude or atmospheric pressure; note the intersection on axis X; join this intersection with the absolute pressure of the compressed air and read the cubic feet of compressed air on the left hand scale

FIG. 2. Nomogram for Converting Volumes of Free Air into Volumes of Compressed Air (Crispell).

Richards' Formula, best for long pipe lines, is:

$$H = \frac{V^2L}{10{,}000D^5a}$$

in which D = diameter of pipe, inches;
L = length of pipe, feet;

V = volume of compressed air delivered, cubic feet per minute;
H = head or difference of pressure required to overcome friction and maintain the flow, pounds;
a = constant for nominal diameter of pipe.

Values of a for Nominal Diameters of Pipe

Inches	Constant	Inches	Constant	Inches	Constant
1	0.350	3	0.730	6	1.000
1¼	0.500*	3½	0.787	8	1.125
1½	0.662*	4	0.840	10	1.200
2	0.565	5	0.934	12	1.260
2½	0.650				

* The values of a for 1¼- and 1½-in. pipe are not consistent with those for other sizes. See footnote on p. 263.

By Richards' formula, the calculated losses of pressure are smaller, and, conversely, the volumes of air discharged are larger, under the same conditions, than those obtained from D'Arcy's formula.

The losses of pressure in a table by F. A. Halsey show that the constants used by him differ materially from those given above. Table IV, containing a series of random examples, shows that in all cases the figures from D'Arcy's formula lie between the others; hence it would appear that the results from this formula are sufficiently accurate for ordinary calculations. Within certain limits, the loss of head increases with the square of the velocity. To obtain the best results the velocity of flow in main air pipes should not exceed 20 or 25 ft. per second (Table V)

TABLE IV.—LOSSES OF PRESSURE IN PIPES AS COMPUTED BY DIFFERENT FORMULAS

Cu.ft. free air transmitted at 75 lb. ($P = 15$ lb.)	Length of pipe, feet	Diameter of pipe, inches	Transmission Losses, Pounds		
			Richards	D'Arcy (Cox)	Halsey
1,000	1,000	4	3.23	3.71	5.02
1,000	1,000	5	0.95	1.17	1.63
1,000	1,000	6	0.35	.46	0.61
4,000	5,000	8	5.92	8.44	13.05
4,000	5,000	10	1.78	2.81	4.20
4,000	5,000	12	0.68	1.06	1.70

TABLE V.*—PRESSURE LOSSES AT DIFFERENT VELOCITIES

Velocity of flow in ft. per sec. Pipe diam., 12 in.	Initial pressure, pounds	Final pressure, pounds	Per cent of initial pressure lost per mile of pipe
25	100	97.6	2.4
50	100	90.6	9.4
100	100	53.8	46.2

* Unwin. Van Nostrand's Science Series, No. 106, p. 78.

When the initial velocity much exceeds 50 ft. per second the loss becomes large; but by using piping large enough to keep down the velocity the friction loss is almost eliminated. For example, in transmitting 875 cu.ft. of free air per minute at an initial pressure of 60 lb., through an 8-in. pipe 7150 ft. long, the average loss including leakage was 2 lb. The velocity in this case was 8⅓ ft. per second. A volume of 500 cu.ft. of free air per min., at 75 lb. gage, can be transmitted through 1000 ft. of 3-in. pipe with a loss of 4.1 lb., whereas if a 5-in. pipe were used the loss would be reduced to 0.21 lb., the velocities being respectively 28 ft. and 10 ft. per second. In driving the Jeddo mining tunnel, at Ebervale, Luzerne County, Pennsylvania, two 3¼-in. drills were used in each heading, with a 6-in. main, the maximum distance of transmission being about 10800 ft. This pipe was so large in proportion to the volume of air required (about 230 cu.ft. free air per minute) that the velocity was only 3⅓ ft. per second. A calculation shows a loss of 0.002 lb., and the gages at each end of the main were found to record practically the same pressure.

Due regard for economy in installation must limit the size of piping, the cost of which in any given case should be considered in relation to the cost of air compression. Diameters of 1 6 in. for the mains are large enough for six to ten drills. Up to a length of 3000 ft. a 4-in. pipe will carry per minute 480 cu.ft. of free air compressed to 82 lb., with a loss of 2 lb. This volume of air will run four 3-in. drills. Under the same conditions a 6-in. pipe, 5000 ft. long, will carry 1100 cu.ft. of free air per minute, or enough for ten drills in constant operation. Branch pipes should not be too small. For a length of, say, 100 ft., a 1¼-in. pipe is small enough for one drill, though 1-in. is often used. Although it is admissible to increase the velocity in short branches considerably beyond 20 ft. per second, extremes should be avoided. To run a 3-in. drill from a 1-in. pipe 100 ft. long requires a velocity of flow of about 55 ft. per second, causing a loss of 10 lb. (see Table VI).

TABLE VI.—AIR DELIVERED BY PIPES OF DIFFERENT DIAMETERS (NORWALK CO.)

Uniform Pressure at the Entrance of the Pipe, 80 Lb. Gage

Nominal size of pipe	1 in.		1¼ in.		1½ in.		2 in.		2½ in.	
Length of pipe in feet	50	100	100	300	100	300	200	500	250	600
Pressures at the Point of Delivery	Cu.ft. of Free Air Delivered as Compressed Air at the Stated Pressures									
79.8	23.2	16.4	35.2	20.3	63.6	36.7	81.7	53.6	142.0	91.7
79.6	33.1	23.4	49.7	28.7	89.9	51.9	119.6	75.7	200.9	129.6
79.4	40.4	28.6	61.0	35.2	109.1	63.0	146.5	92.7	244.4	157.7
79.2	46.8	33.1	70.3	40.6	127.1	73.4	169.1	107.1	283.2	183.1
79.0	52.3	37.0	78.6	45.4	142.0	82.0	189.1	119.7	317.1	204.6
78.8	57.1	40.4	86.1	49.7	155.4	89.7	207.0	131.0	348.4	224.8
78.6	61.6	43.6	93.0	53.7	168.0	97.0	223.3	141.3	377.0	243.9
78.4	65.9	46.6	99.2	57.3	179.3	103.5	238.7	151.1	399.6	258.4
78.2	70.3	49.7	105.4	60.8	190.5	110.0	252.9	160.1	424.1	273.6
78.0	73.7	52.1	110.8	64.0	200.7	115.9	266.5	168.7	446.7	288.6
77.8	77.2	54.6	116.2	67.1	209.9	121.2	279.2	176.7	469.0	302.6
77.6	80.7	57.1	121.4	70.1	219.1	126.5	291.5	184.5	489.6	315.9
77.4	84.0	59.4	126.3	72.9	228.1	131.7	303.4	192.0	509.3	328.6
77.2	87.1	61.6	131.1	75.7	236.7	136.6	314.4	199.0	528.3	340.8
77.0	90.3	63.7	135.4	78.2	245.2	141.6	325.5	206.0	546.5	352.6
76.8	92.9	65.7	139.8	80.7	252.4	145.7	336.1	212.7	564.2	364.0
76.6	95.6	67.7	143.9	83.1	259.8	150.0	346.2	219.1	581.3	375.0
76.4	98.4	69.6	148.1	85.5	267.6	154.5	356.0	225.3	597.5	385.5
76.2	101.0	71.5	152.1	87.8	274.7	158.7	365.6	231.4	613.8	396.0
76.0	103.8	73.4	156.1	90.1	281.3	162.4	375.6	237.3	629.3	406.0
75.8	106.3	75.2	159.7	92.2	288.4	166.6	383.9	243.0	644.5	415.8
75.6	108.7	76.9	163.3	94.3	295.5	170.6	392.8	248.6	659.2	425.3
75.4	111.0	78.5	167.0	96.4	301.7	174.2	401.4	254.0	673.8	434.7
75.2	113.3	80.1	170.4	98.4	307.9	177.8	409.7	259.3	687.8	443.8
75.0	115.5	81.7	173.9	100.4	314.3	181.5	417.9	264.5	701.6	452.7

Compressed-air Piping is of wrought iron, with sleeve couplings or cast-iron flanges into which the ends of the lengths are expanded or screwed; sleeve couplings are used for all except large sizes. The smaller sizes, to 1¼-in., are butt-welded, while all from 1½-in. up are lap-welded to insure necessary strength. Extra heavy piping may be had for high pressures. Wrought-iron spiral-seam riveted, or spiral-weld steel, tubing, sometimes used, is made in lengths of 20 ft. or less. For convenience of transport in remote regions rolled sheets in short lengths may be had, punched around the edges, ready for riveting, and packed four, six or more sheets in a bundle.

All joints in mains and branches should be carefully made. In putting together screw joints see that no white lead or other cementing material is forced into the pipe; it would make ridges and increase the friction loss. Also, each length should be cleaned of all foreign substances which may have lodged inside. For ready inspection and stoppage of leaks, the pipe should, if buried, be carried in boxes sunk just below the surface of the ground; if underground, it should be supported on brackets along the side of the mine workings. Low points in pipe lines form "pockets" for the accumulation of entrained water, and should be avoided, as they obstruct the passage of the air. In long lines, where a uniform grade is impracticable, provision may be made near the end for blowing out the water at intervals, when the air is to be used for pumps or other stationary engines.

For long lines expansion joints are required, especially when on the surface. Underground they are not often necessary, as the temperature is usually nearly constant, except in shafts or tunnels, where there may be considerable variations of temperature between summer and winter.

As each bend or elbow in a pipe line increases resistance, abrupt changes in direction and sharp curves should be avoided. For the same diameter of pipe the resistance due to a bend increases as the radius of the curve diminishes. In the absence of exact data Table VII is given:

TABLE VII. EFFECT OF SHARP BENDS IN PIPE LINES. (Norwalk Co.)

Radius of elbow in terms of diameter of pipe	5	3	2	1½	1¼	1	¾	½
Equivalent length of straight pipe in terms of its diameter	7.85	8.24	9.03	10.36	12.72	17.51	35.09	121.2

These allowances are reasonable, since for steam piping the frictional resistance of an ordinary right-angled elbow is considered equivalent to that due to a length of straight pipe equal to 40 times its diameter. But, the usual bends in wrought-iron air piping are not necessarily so sharp as a standard elbow. When there are many sharp bends the resistance may be very great. This matter should have careful consideration in the stopes of mines, especially when timbered with square sets; as far as possible, the piping should be carried diagonally through the sets, bending the pipe itself where necessary, instead of using right-angled elbows.

Air Hose.—Individual machine drills are connected to the main air pipe by hose, composed of several alternate layers of rubber and canvas. The usual inside diameters are ¾ in. and 1 in. For added strength and protection against injury, the hose is wound with wire or marlin.

The average pressure drop in 50-ft. lengths (the common commercial length) of air hose of different makes is shown by Figs. 3 and 4 (*Min. & Sci. Press.*, Vol. 115, p. 505). These diagrams, based upon tests, emphasize the importance of avoiding the use of hose or pipe of too small diameter. Wire-wound hose wears best, but it is more rigid than that with marlin winding, and may be partly closed by being struck heavily, as by falling rock from blasts. If the winding be too tight, the inner surface of the hose may be corrugated, thus increasing the frictional resistance and drop in pressure. Also, if the hose is coiled between the air main and drills the pressure drop is increased.

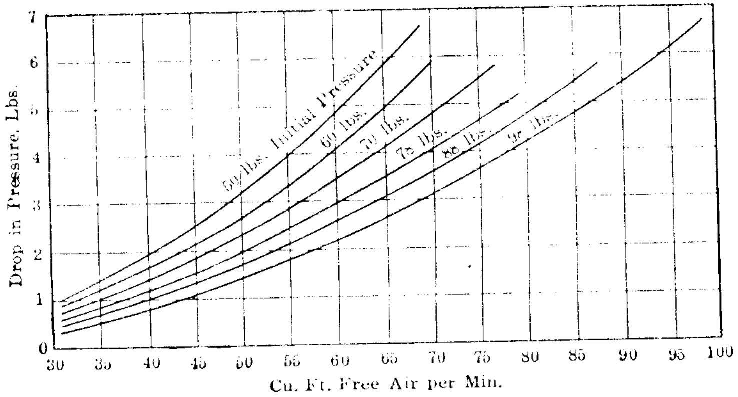

FIG. 3. Drop in Air Pressure in 50 ft. of ¾-in. Hose (W. S. Weeks).

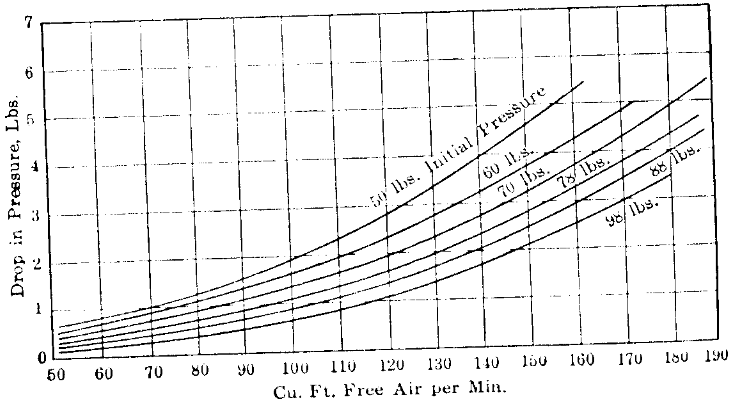

FIG. 4.—Drop in Air Pressure in 50 ft. of 1-in. Hose (W. S. Weeks).

Leakage in Air-transmission Lines should be traced and promptly stopped; it is more expensive than leakage of steam. Air lines are tested from time to time by allowing the air at full pressure to remain in the closed transmission circuit long enough to observe the gage pressure.

Fig. 5 shows an arrangement of apparatus for measuring leakage (*Eng. and Min. Jour.*, Jan. 7, 1922, p. 17). A low-pressure orifice meter (Chapter XXV) is connected to the receiver in the compressor room. The air passing through the meter is admitted by valve *A* to tank *B*, whence it is discharged at orifice *F*. At *E* is a water manometer for observing the air pressure in *B*, the temperature being shown by a thermometer at *C*.

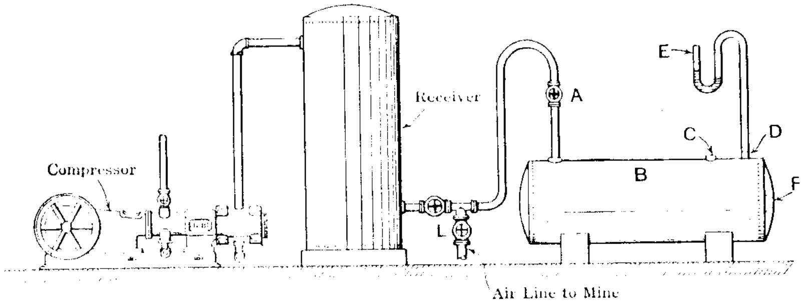

FIG. 5.—Apparatus for Testing Air Lines for Leaks (W. S. Weeks).

When valve *L* to the air line is closed, all the air from the compressor is discharged at *F*. Valve *A* is then adjusted so that the compressor exactly maintains the desired pressure, the meter indicating the corresponding volume of air delivered. Then, on opening valve *L*, the air line and meter are in parallel. Readjust valve *A* until the receiver pressure is again constant at the desired value. The same volume of air is now being delivered as when all was discharged at *F*; but, some air leaks from the air line, the rest going through *F*. A second reading of the meter is then made, and the difference between this and the first reading is the volume of leakage.

CHAPTER XVI

COMPRESSED-AIR ENGINES

COMPRESSED air may be employed as a motive power in an engine in two ways: at full pressure or expansively. By working at full pressure it is understood that the air is admitted to the cylinder throughout practically the entire length of stroke, that is, without cutoff, and that therefore nearly a cylinderful of air at gage pressure is exhausted at each stroke. In this case the work of the air engine is roughly similar to that done in a non-expansive-working steam engine. Among the machines which use air in this way are rock-drills and simple, direct-acting pumps, without rotary parts.

By the term expansive-working it is meant that the air is admitted to the cylinder during only a part of the stroke, and is then cut off and the stroke completed by the expansive force of the air. For operating in this way some equalizing agent, such as the flywheel, is essential, and as a rule a higher initial pressure is employed than when working under full pressure throughout the stroke. It is necessary to distinguish between complete and partial or incomplete expansion. When the air is used with complete expansion the operation in the cylinder is the reverse of adiabatic compression in a compressor, the final pressure being equal to that of the atmosphere. But as air does not undergo condensation, it follows that the lowest terminal pressure in the cylinder must still be sufficiently above atmospheric pressure to produce a proper exhaust, and to overcome the friction of the engine at the end of the stroke. Hence, theoretically complete expansion is impracticable for simple air engines of ordinary design.

Most air engines work with partial or incomplete expansion, the air expanding adiabatically in the latter part of the stroke. The point of cutoff is such that the terminal cylinder pressure exceeds the back-pressure by an amount sufficient to cause a free exhaust. In the conditions here set forth, no reference is made to the thermal changes incident upon adiabatic expansion in the air cylinder. Although, in principle, compressed air is used like steam, both being elastic fluids, there is an essential difference in the results obtained, due to the reduction in temperature. In expanding behind the piston, a given volume of compressed air at a given pressure will not produce the same amount of power as steam under the same conditions. If two curves are constructed, representing the expansion of equal volumes of air and steam, from the same initial pressure down to pressures below that of the atmosphere, it will be seen that the steam pressure at all points of the stroke is considerably higher than the air pressure; and the expansion curve of the air reaches the atmospheric line sooner than the steam curve.

Fig. 1 shows an ideal card, in which the initial pressure is 75 lb., and the cutoff is at $\frac{1}{6}$ stroke. The adiabatic expansion curve of the air shows that the pressure is reduced to zero gage pressure when the air has expanded to $3\frac{3}{4}$ times the initial volume, the mean effective pressure being 18.9 lb. At the end of the stroke the pressure falls to 7 lb. below atmospheric pressure. The steam curve, on the other hand, does not cut the atmospheric line until the expansion reaches $4\frac{1}{3}$ times the initial volume, and the mean effective pressure is 25.2 lb. The lower mean pressure of the air is due to the development of cold during its expansion. The operation is the reverse of compression, and the resulting loss of motive power is analogous to the loss of work in the compressor caused by the generation of heat. Just as the heat of compression reacts upon the air while being compressed in the cylinder, and produces a higher tension than that due to the mere reduction in volume; so conversely, when expansion takes place, the air, which is usually at normal atmospheric temperature on entering the cylinder, rapidly gives up its sensible heat, and the cold reacting upon the expanding air reduces its pressure faster than that which is due to the increase in volume alone. Moreover, this behavior of compressed air is independent of the initial temperature, since the resulting expansion curve would be unaltered. In the case of steam the initial temperature is high, and is reduced but little during expansion from ordinary working pressures down to atmospheric pressure.

A similar comparison may be made for other initial pressures and ratios of cutoff. In every case the mean effective pressure is higher for steam than for air. It follows that, to develop the same amount of power in a given cylinder and with the same initial pressure, the cutoff must be later in the stroke with air than with steam.

So low are the temperatures produced by the expansion of air, from ordinary working pressures of 60 or 70 lb. down to atmospheric pressure, that for a long time the expansive use of compressed air was not considered practicable. In Table I are given the theoretical final temperatures of the exhaust air, in working with complete expansion, and also at full pressure throughout the stroke, for different ratios of initial to final pressure, together with the theoretical efficiencies. The initial temperature is taken at 68° F.*

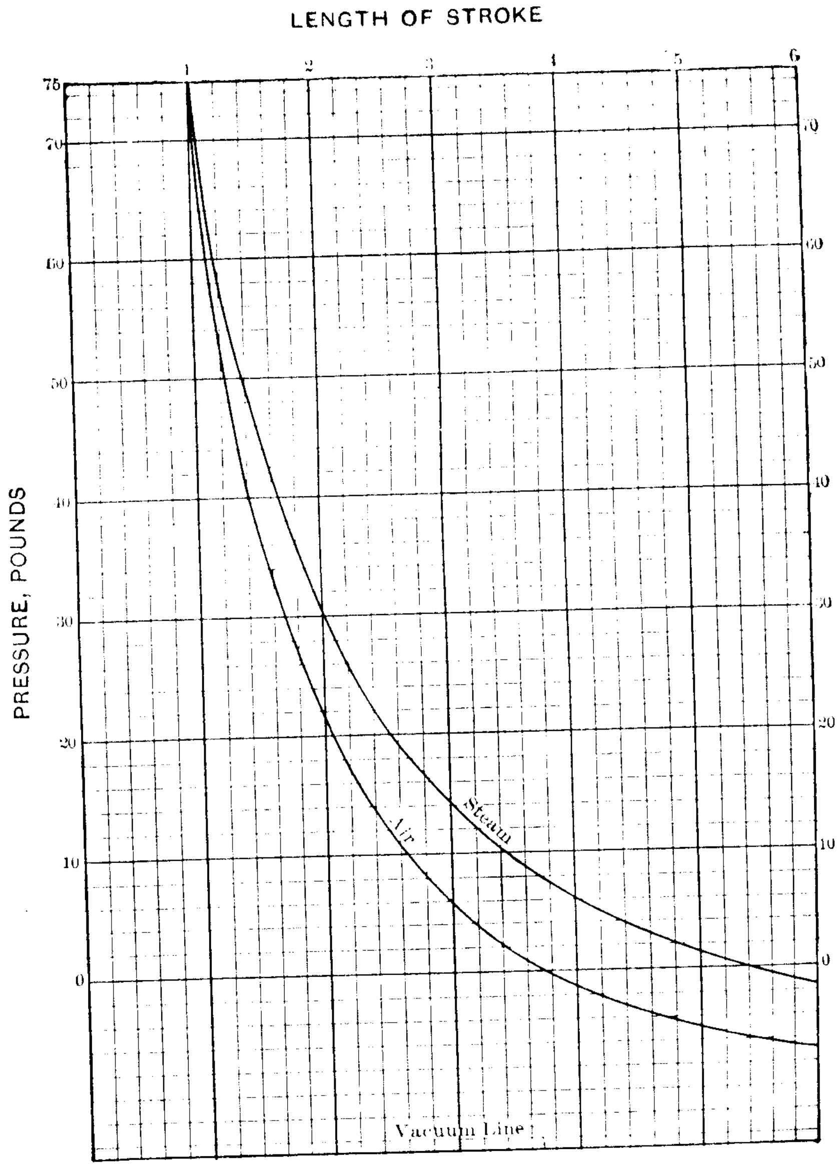

FIG. 1.—Expansion Curves of Steam and Air.

In the table it is shown that by working at full pressure extremely low temperatures of exhaust are avoided; but the efficiency of this method of

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* M. Mallard, " Étude Théorétique sur les Machines à Air Comprimé," p. 27.

TABLE I.—TEMPERATURES OF EXHAUST AIR

Ratio of initial to final pressure	Working with Complete Expansion		Working at Full Pressure	
	Final temperature, deg. F.	Theoretical efficiency	Final temperature, deg. F.	Theoretical efficiency
2	-28.2	0.855	-8.4	0.82
3	76.0	0.806	31.5	0.72
4	-106.6	0.782	-45.7	0.67
5	128.2	0.768	-51.1	0.63
6	-144.4	0.758	59.8	0.60
7	158.8	0.751	-63.4	0.57
8	170.8	0.746	66.1	0.55
9	180.6	0.742	68.0	0.53
10	-189.2	0.739	69.7	0.51

using compressed air is necessarily much below that obtained from expansive working. It is understood that the temperatures here given are theoretical and are never actually reached in practice. The cold produced is modified by several causes: (1) Some heat is transmitted from the external atmosphere through the cylinder walls; (2) the re-compression of the clearance air at each stroke produces heat in the cylinder, to a degree that increases with the initial pressure and the clearance volume; and (3) the presence of even a small quantity of moisture in the air tends in some degree to raise the cylinder temperature.

A few brief notes will here be given concerning the elements of the operation of compressed-air engines, that may be considered more or less applicable for ordinary service, *viz.*, working at full pressure, with partial expansion, or with complete expansion. Isothermal expansion may be neglected, since it involves the application of a sufficient degree of external heat to the air while doing its work in the cylinder to produce a terminal temperature equal to the initial temperature.

1. Working at Full Pressure.—This mode of using compressed air is common for engines like pumps, operating under a constant resistance and not provided with flywheels.

Let P' = the absolute initial pressure of the air;
V' = the initial volume of air, at the pressure P', or K times the volume of 1 lb. of air used per unit of time;
T' = the absolute initial temperature of the compressed air;
T = the absolute final temperature of the air at exhaust, on expanding to atmospheric pressure;

P = pressure of the air at exhaust;
W = foot-pounds of work done.

From the theory of compressed air (Chap. II):
$R = J(C_p - C_v) = 778(0.2375 - 0.1689) = 53.37$, where J is Joule's heat unit, and C_p and C_v are the specific heats of air at constant pressure and constant volume.

As no work is done by the expansive force of the air originally produced by compression, W equals the volume of air used, V', multiplied by the difference between P' and P, or:

$$W = V'(P' - P)$$

Substituting for V' its value, $\frac{KRT'}{P'}$:

$$W = \frac{KRT'}{P'}(P' - P) = 53.37\,KT'\left(1 - \frac{P}{P'}\right)$$

2. Working with Partial Expansion.—The advantages of using compressed air in this way may be obtained from engines possessing flywheels, provided that the cutoff be not too early in the stroke to avoid excessive reduction of cylinder temperature, or else that the air be reheated before entering the cylinder.

In this case the values of P', V', and T' are as above. From the point of cutoff the air expands adiabatically down to a terminal pressure of P'' and volume V'', the final temperature in the cylinder falling to T''. On exhausting, the pressure, volume and temperature become P, V, and T. The work done is composed of three parts:

W' = work between the point of admission and the point of cutoff = $P'V'$;
W'' = work performed by expansion of the volume V' from the point of cutoff to the end of the stroke = $778\,KC_v\,(T' - T'')$;
W''' = negative work due to back-pressure = $-PV''$.

Taking the algebraic sum of these three quantities:

$$W = P'V' + 778KC_v(T' - T'') - PV''$$

But, as under (**1**):

$$V' = \frac{KRT'}{P'} \text{ and } V'' = \frac{KRT''}{P''}$$

Substituting these values of V'' and V''', and for R and C_v their numerical values of 53.37 and 0.1689:

$$W = K\left|53.37T'' + 131.4(T'' - T''') - 53.37T''\left(\frac{P}{P''}\right)\right|$$

$$= 53.37K\left|T'' + 2.46(T'' - T''') - T''\frac{P}{P''}\right|$$

3. Working with Complete Expansion. In the theoretical card, Fig. 2, are shown the relations of the compression and expansion lines, the shaded portion representing the useful work done by the complete expansion of cold air in a motor cylinder. When the expansion is adiabatic, the same relations exist between pressures, volumes, and temperatures as were set forth in the discussion of adiabatic compression:

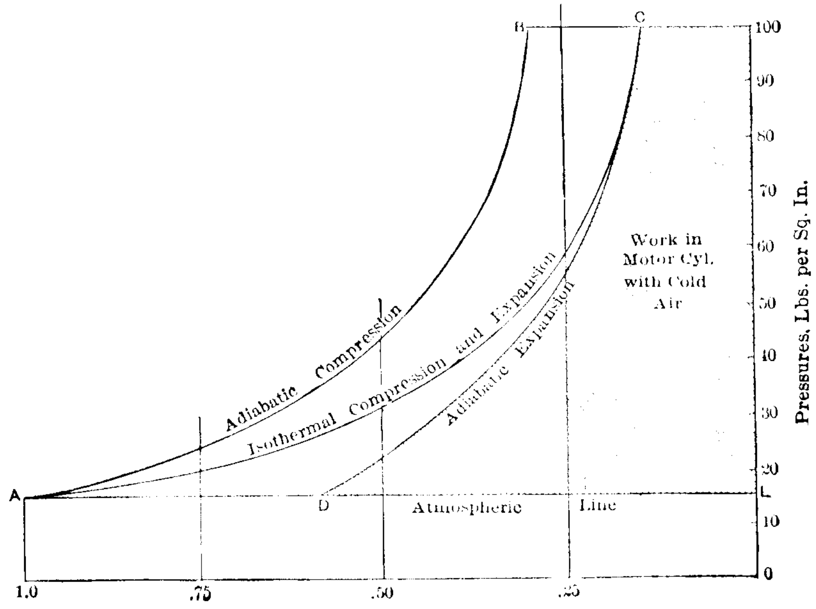

FIG. 2.—Work with Complete Expansion.

$$\frac{P'}{P} = \left(\frac{V}{V'}\right)^{n=1.406} = \left(\frac{T''}{T}\right)^{\frac{n}{n-1}} \quad \frac{n-1}{n} = 0.29$$

The theoretical work done by complete adiabatic expansion may be expressed by a formula like that employed for compression, but with an inversion of certain of the quantities, thus:

$$W = \frac{n}{n-1}PV\left|1 - \left(\frac{P}{P'}\right)^{\frac{n-1}{n}}\right|,$$

in which W = theoretical foot-pounds of work done by the expansion to atmospheric pressure of 1 lb. (13.1 cu.ft.) of free air Substituting the values of the constants, and for working at sea-level:

$$W = 3.463 \times 144 \times 14.7 \times 13.1 \times \left|1 - \left(\frac{14.7}{P'}\right)^{0.29}\right|$$

$$= 96{,}029\left|1 - \left(\frac{14.7}{P'}\right)^{0.29}\right|$$

For example, if P' be 40 lb. gage pressure:

$$W = 96{,}029\left|1 - \left(\frac{14.7}{54.7}\right)^{0.29}\right| = 30{,}440 \text{ ft. lb., or } 2{,}323 \text{ ft.-lb. per cu.ft.}$$

of free air.

Actual Work Done.—In the above expressions no account is taken of the friction of moving parts of the motor engine, or loss of work caused by leakage. In determining the actual work, the general case will be where a cutoff is employed. The relations between initial and terminal pressures and temperatures, for different ratios of expansion in a motor-engine cylinder, are shown in Table II,* the points of cutoff, in tenths of the cylinder stroke, being given in the first column.

The quantities in Table II must be further corrected for piston clearance and the lost volume represented by the air ports and passages of the cylinder, because part of the air expands into these clearance spaces. Therefore, the actual effect of the cutoff, in any given case, is found by dividing the sum of the cutoff plus clearance, by the cylinder volume plus clearance. For example, if the stroke be 10, with a cutoff of $\frac{4}{10}$, and clearance of 6 per cent, the actual volume of the cylinder, including clearance, will be: $(10 \times 0.06) + 10 = 10.6$. Then the sum of the cutoff plus the clearance is $4 + .6 = 4.6$, and the working cutoff becomes $4.6 \div 10.6 = 0.434$. In this manner Table III has been constructed, for use in connection with Table II. It shows the actual cutoff corresponding to the different nominal points of cutoff, for the percentages of piston clearance named at the top of the columns.

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* This table, as well as Table III, is taken in part from those used by G. D. Hiscox, in "Compressed Air, its Production, Uses and Application."

TABLE II.—THEORETICAL RATIOS OF PRESSURES AND TEMPERATURES DUE TO THE EXPANSION OF COMPRESSED AIR IN A MOTOR CYLINDER

Cutoff	Ratio of expansion =1 ÷ cutoff	Ratio of mean to total absolute pressure, for entire stroke	Ratio of mean to total absolute pressure, during expansion only	Ratio of initial to final temperature	Ratio of initial to final absolute temperature, due to expansion only	Ratio of initial to final absolute pressure for ratio of expansion
0.10	10.00	0.249	0.166	0.391	0.513	0.039
.15	6.67	.348	.233	.460	.578	.069
.20	5.00	.436	.295	.518	.627	.104
.25	4.00	.515	.353	.568	.669	.142
.30	3.33	.585	.408	.612	.705	.184
.35	2.86	.647	.460	.652	.737	.228
.40	2.50	.706	.510	.688	.767	.275
.45	2.22	.757	.558	.722	.794	.325
.50	2.00	.802	.604	.754	.818	.378
.55	1.81	.842	.649	.784	.841	.433
.60	1.67	.877	.692	.812	.862	.487
.65	1.54	.907	.734	.839	.882	.545
.70	1.43	.932	.774	.865	.902	.605
.75	1.33	.954	.814	.889	.920	.667

TABLE III.—ACTUAL CUTOFF DUE TO CLEARANCE, FOR THE NOMINAL CUTOFFS IN COLUMN 1

Nominal cutoff	Percentage of Clearance						
	.03	.04	.05	.06	.07	.08	.10
.10	0.126	0.135	0.143	0.151	0.159	0.167	0.182
.15	.175	.184	.191	.198	.206	.213	.227
.20	.223	.231	.238	.245	.252	.259	.273
.25	.272	.279	.286	.293	.299	.305	.318
.30	.320	.327	.333	.340	.346	.352	.364
.35	.368	.376	.380	.387	.392	.398	.409
.40	.417	.423	.429	.434	.439	.444	.455
.45	.465	.471	.477	.481	.486	.490	.500
.50	.514	.519	.524	.528	.533	.537	.546
.55	.564	.568	.571	.576	.580	.585	.591
.60	.612	.615	.619	.623	.626	.630	.637
.65	.660	.664	.667	.670	.673	.676	.682
.70	.709	.711	.714	.717	.720	.722	.727
.75	.758	.760	.762	.764	.766	.768	.772

The theoretical terminal cylinder pressure resulting from adiabatic expansion may be expressed by:

$$\frac{P'}{C^{1.406}} - P,$$

in which C = ratio of expansion = $\frac{1}{\text{point of cutoff}}$ (see column 2, Table II).

For example, for a cutoff at $\frac{1}{10}$ stroke and 65 lb. gage pressure, the terminal pressure (above atmospheric pressure) will be:

$$\frac{65 + 14.7}{2.5^{1.406}} - 14.7 = 7.2 \text{ lb.}$$

The volume corresponding to the nominal cutoff is increased by the clearance, and adds to the mean pressure. Thus, in the above example, assuming the clearance to be 6 per cent, the actual cutoff (Table III) is increased from 0.4 to 0.434, of which the ratio C is $\frac{1}{0.434} = 2.3$. From Table II, column 7, the ratio of initial to terminal pressure, corresponding to the actual cutoff of 0.434, is (by interpolation) 0.31; whence: $(79.7 \times 0.31) - 14.7 = 10$ lb. terminal pressure.

Cylinder Volume Required for a Given Power.—The work per stroke is found by dividing the foot-pounds of work to be done per minute by twice the number of revolutions of the engine (which would be determined for any given size of engine by the ordinary empiric rules of practice). This is substituted, with the initial and final pressures, in the formula for working with full pressure, partial or complete expansion, as the case may be, which is then solved for the initial volume, V', of compressed air used per stroke. To the theoretical cylinder volume thus found, the allowance for piston clearance is added, according to the type of engine. The proper proportion between stroke and diameter of cylinder is finally determined.

The volumes of free air per minute, required for an air engine, per indicated horsepower and for different ratios of cutoff, are shown in Table IV, by F. C. Weber.* The figures given in this table do not include the volume corresponding to piston clearance, which may be found as already shown.

In this table the air is supposed to be used without reheating, and at an initial temperature of 60° F. Reheating will reduce the volume of air proportionally to the ratio $\frac{T_2}{T_3}$, where $T_2 = 459° + 60° = 519°$ F., or absolute temperature; and $T_3 = 459°$ plus the temperature of the reheated air on entering the motor cylinder. Thus, if the air be reheated to 200° F., the above ratio becomes $\frac{519°}{659°} = 0.787$, by which decimal the volume of air as found in the table must be multiplied.

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* *Compressed Air*, Oct., 1896, p. 117.

TABLE IV.—CUBIC FEET OF FREE AIR PER MINUTE USED IN MOTOR ENGINE, PER 1.HP.

Point of cutoff	Gage Pressures, Pounds									
	30	40	50	60	70	80	90	100	110	125
1	23.3	21.3	20.2	19.4	18.8	18.42	18.10	17.8	17.62	17.40
$\frac{3}{4}$	18.7	17.1	16.1	15.47	15.0	14.6	14.35	14.15	13.98	13.78
$\frac{2}{3}$	17.85	16.2	15.2	14.5	14.2	13.75	13.47	13.28	13.08	12.90
$\frac{1}{2}$	16.4	14.5	13.5	12.8	12.3	11.93	11.7	11.48	11.30	11.10
$\frac{1}{3}$	17.5	15.2	12.9	11.85	11.26	10.8	10.5	10.21	10.02	9.78
$\frac{1}{4}$	20.6	15.6	13.4	13.3	11.40	10.72	10.31	10.0	9.75	9.42

So far as mine service is concerned, it has been customary to consider compressed air almost exclusively as an agent for the operation of rock-drills, and in view of its preponderating application to this use its adaptability under proper conditions to the driving of other machines and engines is sometimes overlooked. Of late years, however, with improved methods of compression and reheating, attention has been given to employing compressed air for a greater variety of service; not only underground, but for certain portions of the surface plant of mines as well. Aside from cases where the disposal of exhaust steam would be troublesome, the question is largely one of comparative loss in transmission and the power cost of the air.

Although not strictly in place in this chapter, reference may be made to what has been called the "two-pipe system" or "high-range compressed-air transmission," introduced many years ago by Charles Cummings.*

The machine or engine using the air makes in effect a closed circuit with the compressor. After the air has done its work in the motor cylinder, it is returned to the compressor at the pressure of the exhaust, through a second line of piping. The return pipe connects with a closed chamber at the compressor, in which the inlet valves are placed, thus enabling the compressor to begin its stroke with the cylinder filled under

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* Patent No. 456,911 was issued to Mr. Cummings in 1891.

a considerable initial pressure. Then, after raising the pressure to the original point, the compressor delivers the air into the main, to be used again by the air engine. The actual working pressure of the air engine is, therefore, the difference between the pressures in the delivery and return pipes. Barring leakage, the same air is thus used over and over, the intention being that the compressor shall put back into the air kept in circulation the power expended in the motor-engine cylinder.

Though the compressor itself is not materially different from the ordinary forms, the two-pipe system requires a rather complicated arrangement of piping and valves for charging the apparatus with air at the working pressure adopted, and for governing the speed and output according to the rate of consumption of air.* The advantages of the system are: a higher efficiency than is obtained from moderate-sized compressors of the usual types, and less trouble from freezing at the motor engine by reason of the relative dryness of the air due to its higher tension. The efficiency increases with the pressure employed. In using compressed air without reheating, the two-pipe system is superior in principle to the ordinary mode of operating compressed-air plant. But because of the greater first cost its advantages disappear when reheating can be adopted, and the single-pipe system is then found to be preferable.

The two-pipe system is best suited for machines working at full pressure throughout the stroke, such as machine drills or simple, direct-acting pumps. When the motor works expansively the pulsations become objectionable, as a regular flow of air is not maintained in the return pipe. Under these conditions the inertia and friction of high-pressure air in long pipe lines becomes noticeable and disadvantageous.

As the length of air pipe required for this system is doubled, not only may the first cost of the pipe go far toward offsetting the greater efficiency but, with at least twice as many joints in the pipe lines, the chances of loss from leakage are increased. And if very high pressures are used (pressures of several hundred pounds have been proposed), not only must the piping itself be heavier and more expensive, but the proportionate power loss from leakage is greater. For moderate distances, however, and when working at full pressure under the proper conditions, the foregoing disadvantages may be more than counterbalanced by the superior efficiency of the system. Though not yet in general use, the two-pipe system is said to have given satisfaction at several mines in New Mexico, Colorado, and California, and in 1905 was proposed for use in the Johannesburg gold district. Some prominence is here given to the system because of its novel features and the probability that it may be found useful, if its disadvantages can be overcome. In a paper by H. C. Behr, published in 1905 in the *Transactions of the Mechanical Engineers' Association of the Witwatersrand*, the Cummings system is treated at length, with a discussion of its advantages for air-driven pumps; see also paper by F. Richards in *Power*, Feb. 16, 1915.

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* A detailed illustrated description is given by Frank Richards in *American Machinist*, April 28, 1898, p. 23. See also *Compressed Air Magazine*, Oct., 1907, p. 4599.

Compressed-air Hoists.—In a few mines, compressed-air engines of considerable size are used in deep shafts, where the hoisting is in two stages. The usual case is where an inclined shaft is sunk in the vein, or in the footwall rock, to a point of intersection with a vertical shaft. The compressed-air hoist at the head of the incline delivers ore into pockets at the foot of the vertical shaft, from which the main steam or electric hoist raises it to the surface. This plan is followed at several mines on the Witwatersrand, South Africa, as at one of the Simmer Deep Shafts. For small-scale work standard geared hoists are used, usually without trouble from freezing of moisture, because in intermittent work the cylinders have time to regain normal temperature.

For heavy work the cylinders should be designed for expansive use of the compressed air. The clearance volume is thus reduced, and larger admission and exhaust ports are required, because at the same pressure the density of air exceeds that of steam. Loss in efficiency due to incomplete expansion can be reduced by compounding the cylinders, and, if possible, reheating the air. With a cutoff in the high-pressure cylinder at 0.9 stroke (the minimum practicable starting cutoff), and reheating between the cylinders to the initial temperature, the loss is about one-half that of a simple cylinder, or a saving of 25 per cent of the energy in the air entering the high-pressure cylinder. At the Miami mine, Arizona, the reheating temperature is 350°–375° F.; at the Anaconda mine, 250°–350° F. The volume of air required and the results with different cutoffs are discussed on previous pages (see also Table V).

Small underground compressed-air hoists, portable or semi-portable, are furnished by several makers for sinking shafts from level to level, sinking winzes, raising timbers into position in stopes, temporary haulage of cars, and other light work. They are mounted on a drill column or bar, or can be bolted to a timber or to the floor. Following are examples:

Ingersoll-Rand Co. makes several sizes of "Little Tugger" hoists. They have completely inclosed engines of the square piston type, with four impulses per revolution, so that there is no dead center, and they will run on compressed air or steam. The lightest is the 9H model (Fig. 3). Specifications: weight, without rope, 275 lb.; width, 17 in.; length, $24\frac{3}{4}$ in.; height, $15\frac{3}{8}$ in.; drum, $4\frac{1}{2}$ in. diameter by $4\frac{1}{2}$ in. wide; will carry 300 ft. of $\frac{3}{8}$-in. wire rope. With air at 60 lb., loads of 800 to 1,500 lb. can be raised at speeds of 180 to 40 ft. per min.

TABLE V.—VOLUME OF FREE AIR (60 LB. GAGE) FOR DUPLEX HOISTS

Diameter of cylinder, in.	Stroke, in.	Rev. per min.	Normal Hp.	Actual Hp.	Weight lifted, single rope, lb.	Free air per min., cu.ft.
5.0	6	200	6	11.8	1,000	300
5.	8	160	8	12.6	1,650	320
6.25	8	160	12	19.8	2,500	500
7.	10	125	20	24.2	3,500	604
8.25	10	125	30	33.6	6,000	680
8.50	12	110	40	37.8	8,000	952
10.	12	110	50	52.4	10,000	1,320
12.25	15	100	75	89.2		2,250
14.	18	90	100	125.		3,174

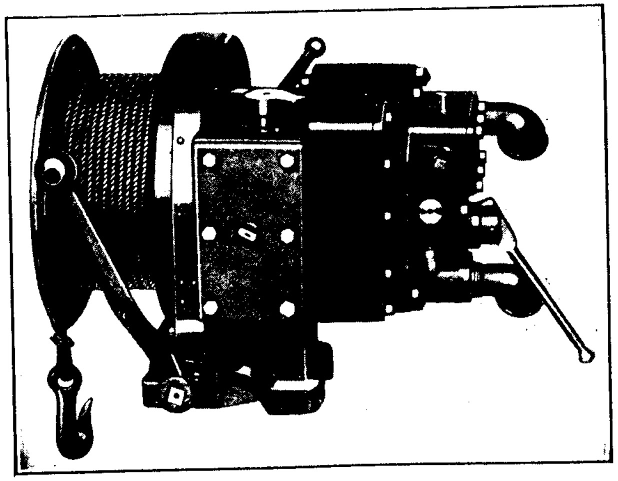

FIG. 3.—Ingersoll-Rand "Little Tugger" Hoist, Model 9 H.

The Sullivan Machinery Co. make a line of five "Turbinair" portable hoists, details of which are given in Table VI. Fig. 4 shows the "HA-3" class. The motor, placed inside the drum and revolving with it, consists of two intermeshing, double helical rotors (Fig. 5), with ball-bearings. The rotors receive compressed air through the hollow drum-shaft, and are inclosed in a casing having very small clearance.

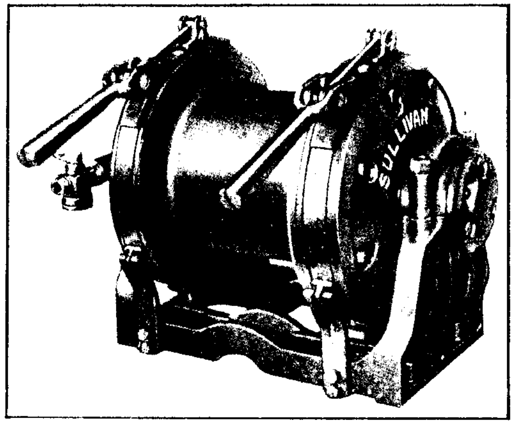

FIG. 4.—Single-drum "Turbinair" Hoist, Class HA-3. (Sullivan Machinery Co.)

Air is admitted to the grooves between adjoining teeth. When a groove (as *A*, Fig. 5) is filled with air, and the rotor has turned far enough to cut off this space from the inlet, the position of groove *A* then corresponds to the cutoff position of the piston of a reciprocating engine. As the rotor continues to revolve, space *A* finally reaches position *B*,

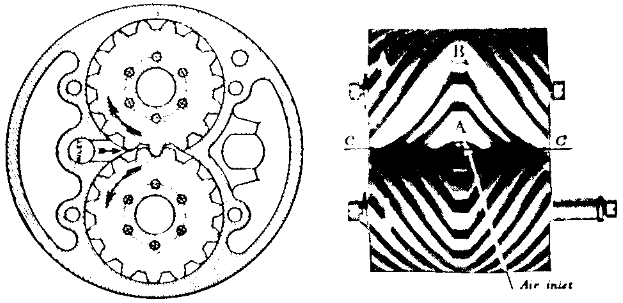

FIG. 5.—Rotors of Sullivan "Turbinair" Motor.

and the air has expanded to fill the space *BA*. Exhaust takes place from the ends of the grooves at the outer edge of the rotors. Since the air acts simultaneously in several of the grooves, an ample starting torque is provided.

Classes HA-2, HA-3 and HS of the Sullivan hoists have one drum; the others, two drums. The first three sizes may be mounted on a column or bar, $4\frac{1}{2}$ in. in diameter, for underground work. The smaller double-drum portable hoists are useful for underground scrapers ("slushers"), for which see Peele's "Mining Engineers Handbook," Second Edition, Sec. 10, Art. 92. Electric drive seems preferable for the larger hoists, from $6\frac{1}{2}$ to 35 hp.

TABLE VI.—SULLIVAN "TURBINAIR" HOISTS

Class	HA-3	HA-2	HDA-2	HS	HDS
Load (lb.), vertical lift, with 50–75 lb. air pressure	2,000	2,000	2,000	2,000	2,000
Lifting speed, 2,000-lb. load (75 lb. air), ft. per min.	110	110	110	110	110
Lifting speed, 2,000-lb. load (50 lb. air), ft. per min.	35	35	35	35	35
Horsepower, at 75 lb. air pressure	$6\frac{1}{2}$	$6\frac{1}{2}$	$6\frac{1}{2}$	$6\frac{1}{2}$	$6\frac{1}{2}$
Net weight, less rope, lb.	345	485	680	475	685
Shipping weight, lb.	410	665	675	555	795
Length overall, in.	24	28	$32\frac{1}{2}$	31	$32\frac{1}{2}$
Width overall, in.	$11\frac{1}{4}$	21	15	15	15
Height overall, in.	18	$22\frac{1}{2}$	$19\frac{1}{4}$	$18\frac{1}{4}$	$19\frac{1}{4}$
Drum diameter, in.	$9\frac{1}{2}$	$9\frac{1}{4}$	$11\frac{1}{8}$	$11\frac{1}{8}$	$11\frac{1}{8}$
Drum length, in.	$9\frac{1}{2}$	$12\frac{1}{2}$	$5\frac{1}{8}$	12	$5\frac{1}{8}$
Capacity of each drum, feet of $\frac{3}{8}$-in. wire rope	150	1,500	160	400	160
Capacity of each drum, feet of $\frac{5}{16}$-in. wire rope	600	2,100	225	550	225

The Chicago Pneumatic Tool Co. make several types of air hoists. Fig. 6 shows class MH 1 single-drum hoist, for mounting on a $4\frac{1}{2}$-in. column for underground service. The motor has a pair of double-acting cylinders, $2\frac{1}{2}$ in. in diameter, by $2\frac{1}{4}$-in. stroke, inclosed in a casing at one end of the drum. Net weight of hoist, without rope, 300 lb.; shipping weight, 380 lb. Drum, 6 in. diameter by 8 in. wide, holds 325 ft. of $\frac{5}{16}$-in. wire rope. Ratio of gearing, 17 to 1. Load capacity: with air at 80 lb., 650 lb. at a speed of 103 ft. per min.; with air at 90 lb., 1,000 lb. at 66 ft. per min.

Portable underground hoists are also made by the Gardner-Denver Co. in two sizes, and by Holman Bros. in three sizes (2, 4 and 6 hp.). The air motor used for the latter is described at the end of this chapter.

Anaconda Copper Co.'s Compressed-air Hoisting Plant.—From 1911 to 1913, a number of the main Anaconda hoists were changed to compressed-air drive by the Nordberg Mfg. Co. Most of the shafts range in depth from about 2,000 to 2,500 ft. The great variation in power required in deep hoisting, and the high peak loads during the period of acceleration suggested the use of compressed air, in connection with an extensive storage system. At one double-compartment shaft, where large quantities of waste rock for filling were being lowered from the surface, the indicated horsepower ranged from −1,600 to +2,300. An elaborate series of tests on the former steam-hoisting plants led to the following plan:

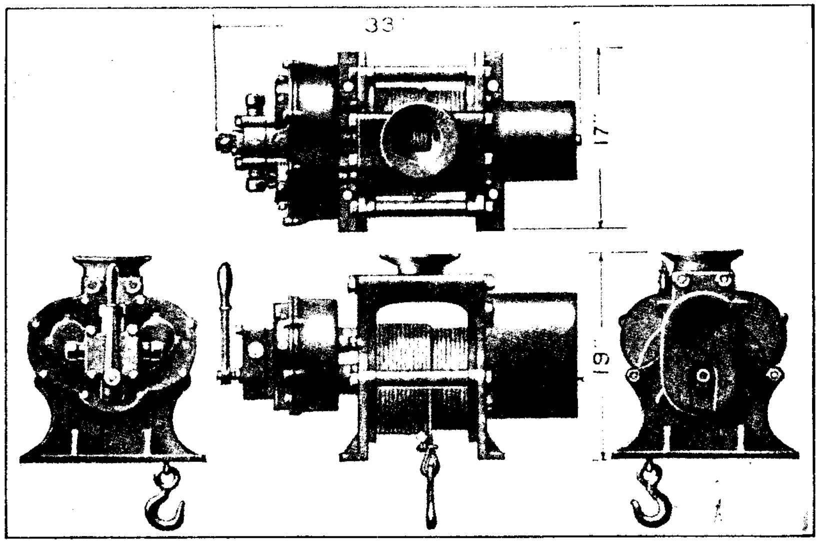

Fig. 6.—Portable Mine Hoist, Chicago Pneumatic Tool Co.

A. An electrically operated compressor plant, with a capacity of 7,650 cu.ft. of free air per minute compressed to 90 lb., the electric current being generated by water power, 150 miles from Butte.

B. Air storage of large capacity. This is connected with the air piping for the rock-drills, so that, in periods when little or no air is used by the hoists, all the air could be turned into the drill mains. To take care of the peak loads, each hoist has several air receivers, aggregating about 8,000 cu.ft. capacity. Maximum air consumption of the largest hoists is 60 cu.ft. of compressed air per second, for a period of 5 sec.; the heaviest unbalanced load requires an average of 40 cu.ft. of compressed air per second for about 1 min., or a total of 2,400 cu.ft. The pipe lines are proportioned for a pressure drop of 5 lb. At times when several large hoists are running simultaneously there is a total draft of 2,600 cu.ft. compressed air per minute, or 22,130 cu.ft. free air (atmospheric pressure at Butte being 12 lb.). The combined maximum consumption of the entire 27 hoists would be 37,200 cu.ft. free air per minute (average,

27,660), the highest peak representing a volume of 44,000 cu.ft. As this measures the minimum required storage capacity to equalize the load on the system, to provide for it, and allowing 10 per cent pressure drop, the total receiver capacity would be 440,000 cu.ft., which is prohibitively large.

A hydrostatic storage plant was therefore installed, consisting of a number of connected air receivers of 66,000 cu.ft. total capacity. About 210 ft. higher than the receivers (corresponding to a pressure of 91.15 lb. per sq.in.) is an open water tank, 100 ft. in diameter, with a pipe leading from its bottom to the receivers. When 66,000 cu.ft. of compressed air have been delivered into the receivers, the water in them is displaced and forced into the upper tank. Hence, the hoists are supplied with air at constant pressure of 91.15 lb., less the pressure drop due to pipe friction.

C. Air reheaters are an important feature of the plant, the heating medium being steam at 200 lb. pressure; reheating temperature, 300° F. Coal used for reheating is $\frac{1}{3}$ lb. per horsepower-hour.

D. The steam hoisting engines, ranging in sizes from 28 in. by 48 in. to 34 in. by 72 in., were fitted with new cylinders, with special valve gearing.*

The advantages of the system are: (*a*) the existing hoists were retained, with comparatively few changes; (*b*) the engines may be run by steam if necessary; (*c*) the air storage is sufficient to operate each hoist for about 20 min., in case of stoppage of the electric power, due to thunder storms, etc.; (*d*) hoisting capacity can be increased by increasing speed, and the same engine will hoist from a greater depth by increasing the compressor capacity; (*e*) part of the energy liberated during the period of retardation, or when waste rock is being lowered, is returned to the power system; (*f*) peak loads on the power plant are eliminated.

Approximate efficiency of the main hoists:

	Per cent
Electric motor efficiency	95
Total efficiency of the compressors	74
Efficiency of hoisting engines	50
Total efficiency of plant, without reheating	35.4
Total efficiency of plant, with reheating	53

Other Compressed-air Hoists have been installed at several mines.

I. Miami Copper Co., Ariz. A 20-in. by 48-in. Nordberg geared hoist, with 10-ft. drums. Air at 80 lb. is reheated to 370° F.; consumption, 2,275 cu.ft. free air per minute, for depth of 675 ft. Rope speed, 750 ft. Capacity, 287 tons per hour. A rough test showed the net shaft horsepower to be about 60 per cent of the indicated horsepower of the compressor. Cost, f.o.b. Milwaukee, $22,500.

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* For further details see: *Trans.* A.I.M.E., Vols. XLII and XLVI; also *Eng. and Min. Jour.*, May 18, 25 and Aug. 3, 1912, and Peele's "Mining Engineers' Handbook," Sec. 12, Art. 10.

II. Franklin Junior mine, Mich., has a combined air and steam hoist. The 10-ton empty skip is lowered unbalanced, and compresses air back into three steam and air storage receivers, each 10 ft. in diameter by 32 ft. long. Air cylinders, 36 in. in diameter, are tandem to 46-in. steam cylinders; stroke, 72 in.; Corliss valve gear. Drum, 15 ft. in diameter, holds 5,130 ft. of $1\frac{5}{8}$-in. rope. Two 200-hp. boilers are used. When hoisting, the ends of the air cylinders are by-passed, and no air is compressed; when lowering, the steam-cylinder exhaust valves are open, and the air compressed in the air cylinders is discharged into the receivers. A reducing valve keeps the receiver pressure at 75 lb. The skip is lowered with the receivers full of steam at 75 lb.; the compressed air raises the pressure to 95 lb., which is then available to start hoisting. When the pressure falls below 75 lb., steam is admitted to complete the hoist.

III. Copper Queen mine, Arizona, has three direct-acting and two geared hoists operated by compressed air. The larger direct-acting hoists use 1,100–1,600 cu.ft. free air per shaft horsepower.

Hele-Shaw-Beacham Air Motor (Holman Bros). This unique air engine, brought out in 1927, is similar in design and general principle of operation to the single-acting rotary compressor of the same make (Chapter III). There are three single-acting cylinders, set radially. The following standard sizes are now being built:

Cylinders, in.		Rev. per min.	Air pressure, lb.	Brake hp.	Weight, lb.	Overall dimensions, in.		
Stroke	Diam.					Width	Length	Height
$1\frac{1}{2}$	5	1,500	80	15	240	$17\frac{1}{2}$	22	21
$2\frac{3}{16}$	$8\frac{3}{8}$	1,000	60	33	812	$22\frac{1}{2}$	$30\frac{11}{16}$	$24\frac{3}{8}$

The smaller motor is shown in Fig. 7. A third size, developing 23 hp. at 1,000 rev. per min. and 40 hp. at 2,000 rev. per min., is now (1929) being designed. Due to their compactness and light weight, these motors can readily be moved from place to place, and are therefore well suited to underground service in mines, as for driving centrifugal pumps, winze-hoists and light haulages. They are reversible, will run at any speed from zero to 2,000 rev. per min., and weigh less and occupy less space than electric motors of the same power. The design of the rotating cylinder block (with which the cylinders are integral), the single central valve, and the mode of producing the reciprocations of the pistons by "floating rings," will be understood by reference to the description accompanying Fig. 51, Chapter III.

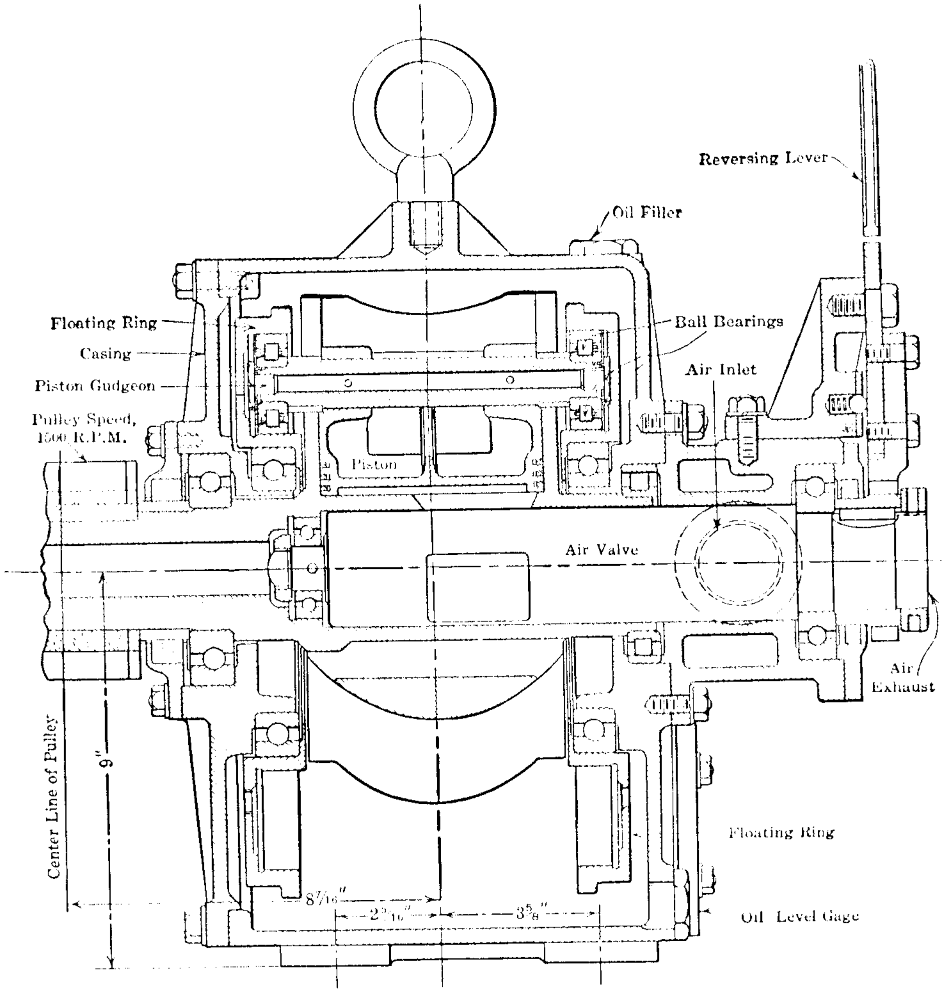

Fig. 7.—Hele-Shaw-Beacham Compressed-air Motor (Holman Bros.).

On the outward or power stroke, the compressed air is cut off at about $\frac{3}{4}$ of the stroke and allowed to expand until the exhaust port opens. Toward the end of the return stroke, after the exhaust has closed, the air remaining in the cylinder is compressed into the clearance spaces in the dished piston and cylinder port, so that these are filled with compressed air when the inlet port opens. To reverse the motor, the central valve is shifted by the hand lever, thus placing the inlet and discharge ports on the opposite side of the vertical center line, but in the same relative positions to the top and bottom centers as before. When the reverse lever is in its central position the valve ports are blanked by the valve chest, and the motor stops.

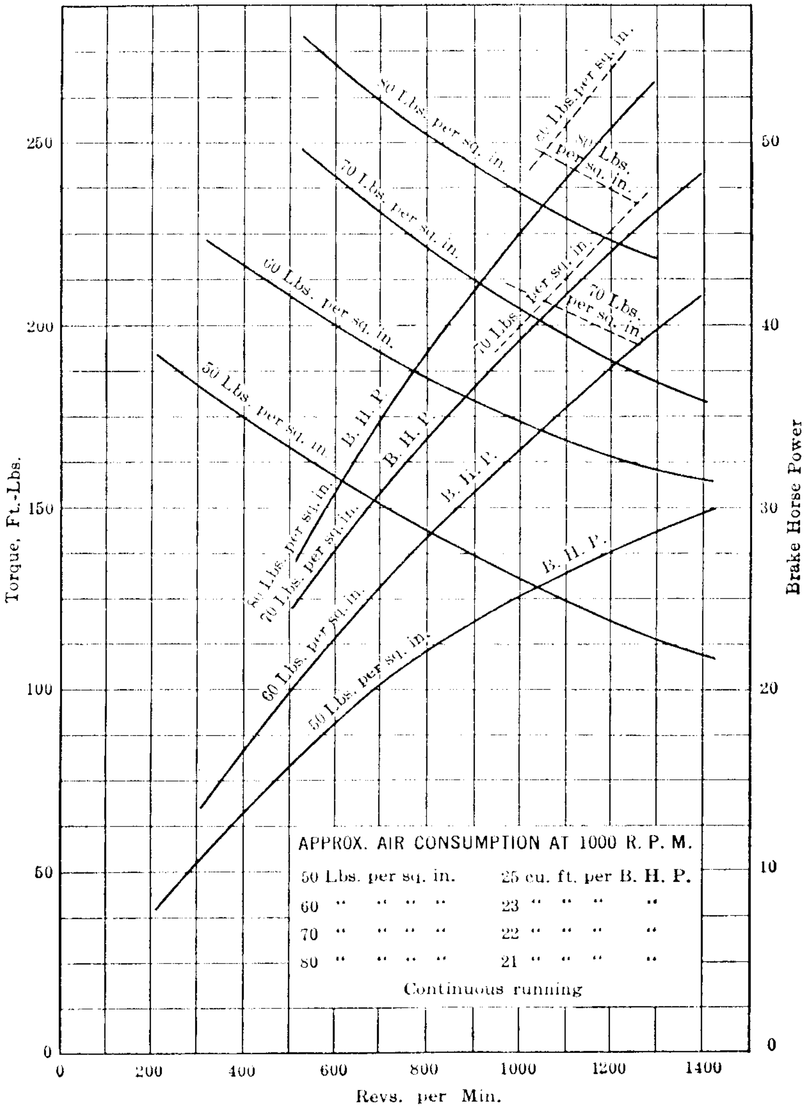

Fig. 8.—Hele-Shaw-Beacham Air Motor. Brake Hp. and Torque of the $2\frac{3}{16}$ by $8\frac{3}{8}$-in. Size.

The results of a test of the 33-hp. motor, with different air pressures, are shown in Fig. 8.

Holman Bros. have applied the Hele-Shaw-Beacham motor to driving small, portable hoists. These may be mounted on a column or stretcher-bar or bolted to a foundation, or supported on a turn table. The motor is placed inside of the drum, which is carried on ball-bearings on the outside of the motor casing. Two sizes are made: $6\frac{1}{2}$ and 15 hp., at 80 lb. air pressure, the load raised depending upon the ratio of gearing. The larger hoist has a drum $6\frac{1}{4}$ in. wide by $15\frac{1}{2}$ in. in diameter, holding 450 ft. of $\frac{1}{2}$-in. rope or 620 ft. of $\frac{3}{8}$-in. rope. It is fitted with reduction gearing of 28.9 to 1, giving a rope speed of 250 ft per min. at 1,500 rev. per min. of the motor. Lifting capacity of the smaller hoist is 2,000 lb. at 110 ft. per min.

These hoists have band brake, gear-disengaging lever, and control lever operating the central air valve, as in Fig. 7. As the control lever is spring-loaded, it returns automatically to the stop position when released. The hoists are suitable for winze work, or for handling timber in stopes.

CHAPTER XVII

FREEZING OF MOISTURE DEPOSITED FROM COMPRESSED AIR

The moisture present in the atmosphere at all times in greater or less quantity is carried with the air discharged from the compressor. Part of the water is deposited in the air receiver, but some of it passes unavoidably into the air main.

The capacity of air for carrying moisture varies with its temperature. Under ordinary atmospheric conditions 1,000 cu.ft. of air contain about 1 lb. of water. When its volume is reduced in the compressor cylinder, the heat of compression augments its moisture-carrying capacity. Any subsequent decrease in temperature reduces this capacity, and if the air is saturated the excess of moisture is deposited. Volume for volume, the capacity of air for moisture is independent of its pressure or density. That is, at the same temperature, a cubic foot of air at atmospheric pressure will hold in suspension the same weight of water as a cubic foot at 100 lb. pressure. But the effect of this must not be misunderstood. If a certain volume of saturated atmospheric air is compressed isothermally to $\frac{1}{10}$ of its original volume, its water capacity is also reduced to $\frac{1}{10}$, and $\frac{9}{10}$ of the water originally present is deposited. Therefore, while the capacity of air for carrying moisture varies with the temperature, it changes also with any increase or decrease of pressure which changes its volume.

Relative Humidity, or degree of saturation, of atmospheric air is the ratio of the weight of watery vapor present to that corresponding to saturation. It is determined by the hygrometer (psychrometer), consisting of two thermometers, one an ordinary "dry-bulb" thermometer, the other a "wet-bulb," the bulb of which has a cover of wetted thin muslin. Evaporation caused by a current of air, or by "whirling" or swinging the hygrometer, lowers the reading of the wet-bulb thermometer in geometric proportion to the decrease in relative humidity of the air.

The formula for determining relative humidity, as used by the U. S. Weather Bureau, is:

$$e = e' - 0.000367P(t - t')\left(1 + \frac{t' - 32}{1571}\right),$$

where t and t' are the temperatures indicated by the dry- and wet-bulb thermometers, P the corrected barometric reading in inches, e' the maximum or saturation pressure of aqueous vapor at temperature t' of the wet-bulb, and e the vapor pressure corresponding to the observed temperatures t and t'. In swinging the hygrometer, its velocity should be not less than 15 ft. per second.

The extensive tables of the U. S. Weather Bureau are published in their Bulletin No. 235. For compressed-air calculations the condensed Table I will be found sufficient.

TABLE I. HYGROMETER READINGS

	Difference of temperatures, $t - t'$																	
Air temp.	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18
	Degrees of humidity, saturation being 100																	
32° F.	87	75																
42	92	85	78	72	66	60	54	49	44	40	36	33	30	27				
52	93	86	80	74	69	64	59	54	50	46	42	39	36	33	30	27	25	
62	94	88	82	77	72	67	62	58	54	50	47	44	41	38	35	32	30	28
72	94	89	84	79	74	69	65	61	57	54	51	48	45	42	39	36	34	32
82	95	90	85	80	76	72	68	64	60	57	54	51	48	45	42	40	38	35
92	95	90	85	81	77	73	70	66	62	59	58	53	50	47	45	43	41	38

The dry-air value of R (Chapter II) is commonly used for the entire cycle of compression and use of the air. But there is a loss of work in compressing the watery vapor, because most of it is deposited as water in the intercooler, receiver and pipe-line, before the air reaches the machines using it. For example, if 1,000 cu.ft. of air enter a single-stage compressor at 80° F. and 70 per cent humidity, the vapor will occupy 34.3 cu.ft., representing a volumetric loss from this cause of $0.7 \times 34.3 = 24$ cu.ft., or 2.4 per cent of the total intake volume. In two-stage compression, the effect of the intercooler and additional jackets reduces this loss to about 2 per cent. If the air enters at say 100° F., which may occur if it is taken directly from the hot compressor room, as is often the case, the lost work will be roughly 5 per cent for single-stage and 4 per cent for two-stage compression.

The volume of vapor contained in atmospheric air can be computed by the formula:*

$$V_s = \frac{Vp_s}{p},$$

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* A. Hofmann, *Power*, Jan. 13, 1914 and *Eng. and Min. Jour.*, Feb. 14, 1914, p. 366.

where V_s = cubic feet of vapor only at pressure p;
V = cubic feet of air and vapor at pressure p;
p = absolute pressure per square inch of air and vapor;
p_s = absolute pressure per square inch, exerted by the vapor only.

If the mixture is saturated, p_s is the pressure of saturated steam, as given in steam tables for the temperature under consideration. The weight of the vapor present = $V \times D$, where D is its weight per cubic foot; and, if the air is saturated, D is also the weight per cubic foot of saturated steam. Fig. 1 shows the relations of these factors.

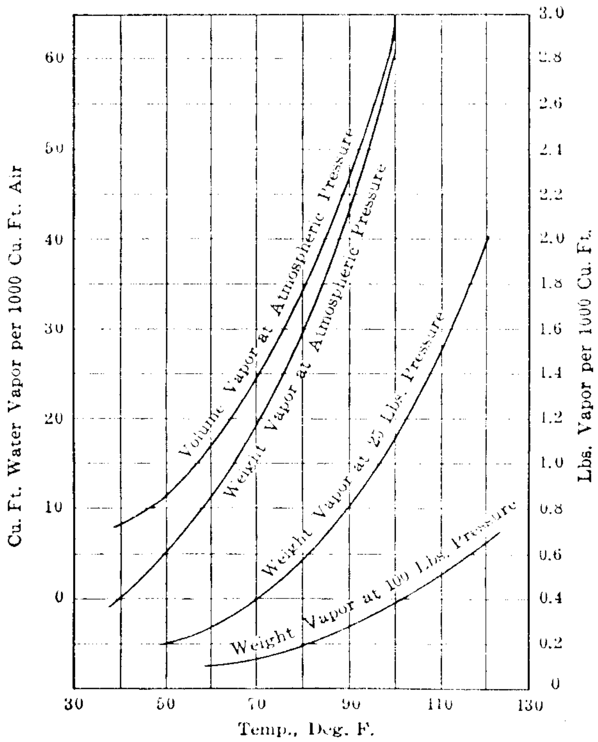

Fig. 1.—Vapor in Air at Different Pressures and Temperatures (*Power*).

Causes of Freezing.—Certain conditions are required to cause freezing of compressed air: deposited moisture must be present, and it must be subjected to a temperature below the freezing-point. So long as the temperature does not fall low enough, the presence of moisture can do no harm. Although one of the recognized functions of the air receiver is to permit the deposition of water before the air passes into the pipes, still, unless the receiver is extremely large, the air leaves it warm

usually even quite hot—and therefore carries with it considerable moisture. In the case of wet compressors, unless liberal sprays are used to attain effective cooling, the air is apt to contain more moisture than that from dry compressors. A well-designed injection compressor, however, not too small for its work and therefore running at a moderate speed, will deliver cool air which will not give trouble from freezing. The air having attained nearly normal temperature before entering the pipe-line, its moisture-carrying capacity undergoes but little further reduction while passing through the pipe, and only a small amount of additional deposition takes place. With dry compression the percentage of humidity of the intake air, and the temperature at discharge, determine the quantity of water carried out of the cylinder. The humidity, in turn, varies with the weather. Changes in the weather may quickly be followed by variations in the quantity of moisture deposited in the receiver and pipe-line. When the air is finally expanded in doing its work in the air engine, intense cold is produced as the pressure falls, and the latent heat of compression is absorbed. It is here that the moisture carried with the air into the pipes makes its appearance as frost and causes trouble. Watery vapor itself, depositing a light, snow-like frost, does not tend to clog the air passages and ports as much as entrained water in a finely divided state, which will gradually form accumulations of solid ice and choke the exhaust wholly or in part.

Prevention of Freezing. The difficulties which may arise from the conditions just outlined are apt to be exaggerated. That freezing not infrequently occurs is true, but with a properly designed and arranged plant it may easily be avoided. Two things require attention: first, the air should be caused to drop its moisture as completely as possible before entering the main; second, provision should be made for draining off what deposited moisture remains in the pipe-line, before the air passes to the machine in which it is to be used. Although this is a simple matter, the means for accomplishing it are often neglected. Considerable quantities of water may collect in low places in the pipe-line and, if not blown out at intervals, will be carried into the ports, cylinders, and exhaust passage of the air machine and there freeze.

Granting that the air leaves the receiver near the compressor practically saturated and still warm, it is evident that a great improvement in working conditions may be realized by introducing a second receiver as close as possible to the machines using the air. In mining, the second receiver is placed underground (Chapter IX). Before reaching it, the temperature of the air will have become normal, and the entrained moisture from the pipe-line may readily be trapped and drawn off. It may be remarked that automatic water-traps are preferable to valves or cocks for getting rid of the water. As a rule, when the compressed air is to be used expansively, a special aftercooler should be introduced, placed as close as possible to the compressor. In any case, the receiver should be of ample size to insure the deposition of the moisture. The advantages of reheating the air before use will be taken up later.

Influence upon Freezing of High Pressures in Transmission.—The statements made in the first part of this chapter suggest an important consideration, *viz.*, in transmitting power by air at a high pressure there is less liability to trouble from freezing than when low pressures are employed, provided that the length of pipe-line is sufficient to allow the air to be completely cooled and drained of its water while still under high pressure. At a low pressure a greater volume of air is required to furnish a given amount of power than when at a high pressure. More moisture must, therefore, be dealt with, and at a low pressure it can not be so thoroughly separated before the air is used. Suppose the transmission to be at a high pressure, and through a pipe long enough to allow the air to reach normal temperature. If the deposited moisture is drained away while the air is at its maximum pressure; then, if the air be subsequently expanded down to a lower pressure suitable for working (with a corresponding increase of volume) and allowed to regain its normal temperature, the percentage of moisture will be reduced, so that the air may be relatively very dry. When finally used in the air engine there will not be enough moisture present to cause troublesome freezing.

Deposition of Moisture by Reducing Pressure.—To carry out the above plan, the underground receiver being placed close to the pump or other air engine, the pressure in the air main is reduced by "wire-drawing." This is done by a reducing valve set in the main at a point several hundred feet before reaching the auxiliary receiver. At the lower pressure, the temperature of the air falls, so that most of the moisture is deposited and collects in the receiver. The proper amount of reduction obviously depends upon the relation of the transmission pressure to that required for operating the air engine. If the latter has compound cylinders, still further improvement may be effected by admitting a little "live air" into the exhaust pipe from the high- to the low-pressure cylinder. During expansion in the low-pressure cylinder, the warmth of the live air raises the cylinder temperature, so there is no deposition of frost in the final exhaust passages.*

Protection of Surface Piping. In using compressed air, even for mining purposes, it often becomes necessary to carry lines of air pipe considerable distances on the surface. To prevent condensation and freezing of the moisture in winter by external cold, all surface piping

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* In this connection see *Colliery Guardian*, Oct. 30, 1896, p. 821.

must be protected. If exposed to temperatures below the freezing-point, the inside of the pipe will become coated with ice and its effective cross-section reduced. A serious diminution of area may thus be caused at low points in the pipe-line, where water tends to collect; or the pipe may even be frozen solid in such places by the gradual accumulation of ice. Underground the temperature is rarely, if ever, low enough to render any protection necessary, except in cold, down-cast shafts, or in tunnels in which there is a strong inward draught.

Some time ago, at one of the Butte copper mines, a simple and inexpensive device was employed to prevent the freezing of moisture in a long line of surface piping. The air main of a large compressor plant was carried on the surface some hundreds of feet before reaching the shaft. During the winter months it was at times difficult to get sufficient air pressure in the mine because of the partial choking up of the pipe. As the volume of compressed air was too large to be dealt with by the ordinary receiver, a series of old tubular boilers were placed close to the compressor house. The hot air, at 80 lb. gage pressure, in passing through the receivers, from one to another, was cooled down practically to atmospheric temperature and as a consequence a large part of its moisture was deposited. It was found that discarded tubular boilers, when strong enough, were well suited to this purpose, because of the large surface presented to the cold outside air; especially when they are set horizontally, so that there is a free circulation of air through the tubes. A blower might be used for the same purpose in a warm climate, or the boilers submerged in cold water. This effectual remedy is worthy of adoption where the conditions are similar.

CHAPTER XVIII

REHEATING COMPRESSED AIR

It has been shown (Chapter II) that, due to the heat produced in the cylinder, the compression line of the air card of an ordinary compressor closely approaches the adiabatic line. This involves the expenditure of more work than if the compression were isothermal. But, after the warm air from the compressor passes into the pipe-line, its temperature is soon reduced to that of the surrounding atmosphere, resulting in loss of pressure or volume. If the air could be utilized immediately after leaving the compressor, the loss of work would be avoided; a condition that is rarely practicable.

By reheating the air the lost work can be recovered, provided the reheater be so placed that the air from it passes at once into the cylinder of the machine using it. Reheating is readily applied and serves two purposes: it increases the volume of the air and eliminates trouble from freezing of moisture in the exhaust ports of the rock-drill or other air engine.

Results of Reheating are shown by comparing the heat units required to produce a given volume of air at a given pressure in a compressor cylinder with the heat units for accomplishing the same result by expanding the air by the direct application of heat. It is evident from the following theoretical calculation * that an added volume of air is obtained at a lower power cost than if this air were produced in the cylinder itself:

Weight of 1 cu.ft. of steam, at 75 lb. gage = 0.2089 lb.

Total heat units in 1 lb. of steam, at 75 lb., produced from water at 60° F. = 1151.

Total heat units in 1 cu.ft. of steam at 75 lb. = 1151 × 0.2089 = 240.44.

To produce by compression in a steam-actuated air compressor 1 cu.ft. of compressed air at 75 lb. gage and 60° F., about 2 cu.ft. of steam at the same pressure are required,† making the thermal cost of 1 cu.ft. of compressed air, at the above temperature and pressure, 240.44 × 2 = 480.88 heat units. The air is here supposed to have lost its heat of compression by being stored or transmitted to a distance, so that the 480.88 heat units represent its cost at the motor where it is used.

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* Frank Richards, "Compressed Air Practice."

† That is, the efficiency of the compressor is assumed to be 50 per cent.

Result of reheating:

Weight of 1 cu.ft. of air at 75 lb. and 60° F. = 0.456 lb.

Units of heat required to double the volume of 1 lb. of free air at 60° F. = 123.84.

Units of heat required to double the volume of 1 cu.ft. of compressed air at the above temperature and pressure = 123.84 × 0.456 = 56.47.

Comparing the thermal cost of 1 cu.ft. of air compressed in a cylinder with that of 1 cu.ft. obtained by reheating:

$$480.88 : 56.47 :: 1 : 0.1174$$

that is, the cost in heat units of a volume of air produced by reheating is less than ⅛ of that required to produce the same volume by compression.

This theoretical result cannot be attained in practice. To effect such a saving a perfect reheater would be necessary, and the air after reheating would have to pass directly into the motor cylinder. A farther conveyance of the air in pipes for even a short distance rapidly lowers its temperature and pressure.

Reheating Temperatures.—At constant pressure the volume of air is proportional to its absolute temperature, or 459° F. plus the sensible temperature above zero. The absolute temperature of air at 70° F. is 459 + 70 = 529°. In doubling the volume by the application of heat the absolute temperature becomes 1058°, and 1058 − 459 = 599°, which is the corresponding thermometric temperature. But this temperature is greatly reduced by the time the air reaches the motor cylinder, and still more heat is lost in the cylinder before its work is done. To reheat the air to a temperature which would double its volume in the motor cylinder would require reheating to a temperature much higher than 599°. If the temperature is raised by the reheater to 400° F. a loss of, say, 100° should be allowed for cooling before the air is actually used. The absolute cylinder temperature is then 300 + 459 = 759°, and the added volume of compressed air practically available is:

$$529 : 759 :: 1 : 1.43 +.$$

That is, by reheating to 400° F., there has been an effective increase of about 43 per cent in the volume of compressed air. For proper lubrication, a higher temperature would be undesirable in the motor cylinder, and no material increase in economy could be realized in the operation of the motor. In practice the gain from reheating is considerably less than is here shown. For machine-drills and small, single-cylinder pumps, taking air at nearly full stroke, the increase of work ranges from 30 to 35 per cent, without deducting the cost of the fuel used in the reheater. Expansive-working engines show a higher efficiency.

For some purposes the determination of the quantity of heat to be imparted in reheating is based on the temperature at which the air leaves the compressor cylinder, the idea being to recover the heat subsequently lost in cooling. Suppose, for example, that the compression is practically adiabatic, as is usual in single-stage compressors. Taking as the unit 1 lb. of air, or 13.2 cu.ft., at 65° F., and compressing to 70 lb. gage, the heat of compression is:

$$T' = T\left(\frac{P'}{P}\right)^{0.29} = 65° + 459°\left(\frac{70 + 14.7}{14.7}\right)^{0.29} = 869° \text{ absolute,}$$

and the final thermometric temperature is, 869° − 459° = 410° F. The rise in temperature due to compression is therefore:

$$410° - 65° = 345° \text{ F.}$$

If the compressed air be subsequently cooled to 65°, its volume becomes: $\frac{14.7 \times 13.2}{84.7} = 2.29$ cu.ft. In using this air without reheating and non-expansively, in a rock-drill having 10 per cent clearance, the work done is:

$W = (2.29 \times 144 \times 84.7 \times 0.9) - (2.29 \times 144 \times 14.7) = 20{,}290$ ft.-lb.

But if the air be reheated to the final temperature of compression (345° F.), the work is:

$W' = \frac{869°}{524°} \times 20{,}290 = 33{,}478$ ft.-lb., and the gain by reheating is therefore: 33,478 − 20,290 = 13,188 ft.-lb., or 39 per cent.

The thermal cost of reheating this air will be: 345° × 0.2375 (specific heat of air at constant pressure) = 81.9 thermal units (B.t.u.), equivalent to 81.9 × 778 = 63,718 ft.-lb. of work. Hence the efficiency of reheating in this case is:

$$13{,}188 \div 63{,}718 = 20.7 \text{ per cent.}$$

A working test, by Prof. Alex. B. W. Kennedy, on a reheater supplying air for a 10-hp. motor, gave the following results: The air was reheated to 315° F., by burning about 0.39 lb. coke per indicated horsepower-hour, producing an increase of about 42 per cent in the volume, and, if the efficiency had remained the same as during the trials with cold air, there should have been a decrease of air consumption in the ratio 1 ÷ 1.42 = 0.7. The volume of cold air used (admission temperature, 83° F.) was 890 cu.ft. per horsepower-hour; the volume when reheated was 665 cu.ft., or 75 per cent; so that the full economy from reheating was nearly realized.

A summary of the results from two experiments on the above plant with cold, and two with reheated, air show:*

1. With cold air. Incomplete expansion, wire-drawing, and other such causes, reduced the actual horsepower of the motor from 0.50 to 0.39.

2. By heating the air to about 320° F. the horsepower at the motor was increased to 0.54. The ratio of gain due to reheating was therefore 0.54 ÷ 0.39 = 1.38.

3. Deducting the value of 0.39 lb. coke per horsepower-hour, used in heating the air, the net efficiency becomes 0.47, instead of 0.54, and the ratio of gain is reduced to 0.47 ÷ 0.39 = 1.205.

These experiments, though not made with a well-designed reheater, show a substantial net gain from reheating.

Where reheating is employed in mines, the added temperature is usually much less than that indicated above. In common practice, the air is reheated to about 250° F., which produces an increased volume of 30 to 35 per cent. Good results may be obtained by the application of even less than 100° F.

The results of experiments by Riedler and Gutermuth, on the consumption of reheated air by a single-cylinder Corliss 80-hp. engine, are given in Table I. Though originally designed as a steam engine, the only change made for its work with compressed air was to jacket the cylinder by the reheated air on its way to the valve chest. The initial pressure was 80 lb. gage and the temperature of the air in the reheater did not exceed 338° F., at a coke consumption of 0.176 lb. per horsepower-hour.

Table I

Test	Temperature of air		Consumption free air per hp.-hr. in cu.ft.	Indicated horsepower	Efficiency
	Admission, deg. F.	Discharge, deg. F.			
1	264.2	69.8	462.77	72.3	0.89
2	305.6	84.4	431.09	72.3	0.90
3	320.0	95.0	418.55	72.3	0.91
4	338.0	120.2	432.12	65.0	0.87

Fig. 1 is a convenient diagram showing graphically the increase of volume of compressed air, at constant pressure, due to reheating from the stated initial temperatures by the application of different degrees of heat. The relative volume at any temperature of reheat is read off on

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* "Experiments upon the Transmission of Power by Compressed Air in Paris." Van Nostrand's Science Series, No. 106, p. 35.

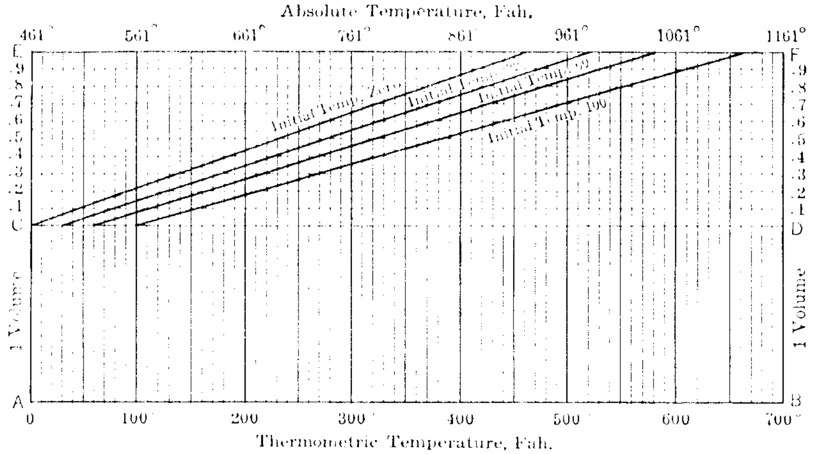

Fig. 1.—Increase of Volume by Reheating (Richards, "Compressed Air Practice").

the vertical lines. If the original volume be represented by AC, the height of each horizontal line above CD indicates an added one-tenth volume. At the horizontal line EF, the volume is doubled.

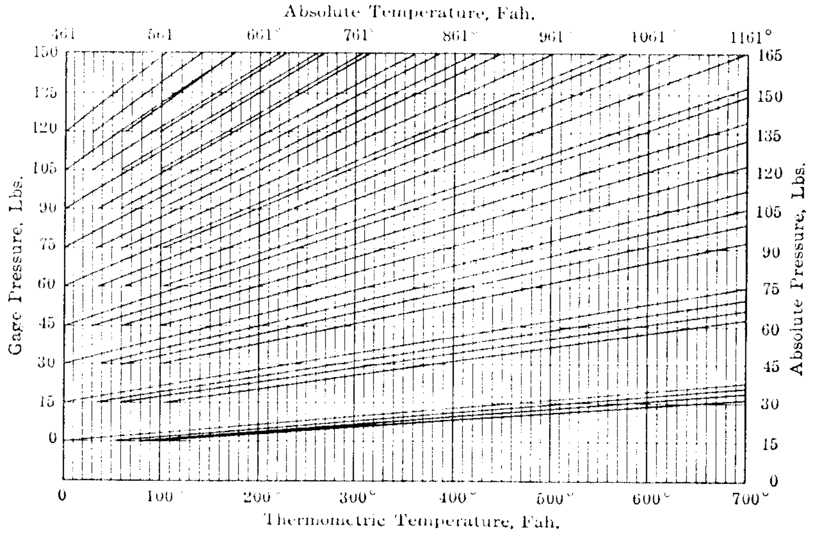

Fig. 2.—Increase of Pressure by Reheating (Richards, "Compressed Air Practice").

Fig. 2 shows the increase of pressure due to reheating, the volume being constant. The results are read from the diagonal lines, when the

air is heated as in the previous diagram, and from the same initial temperatures. The vertical distances between the successive horizontal lines approximately represent atmospheres, the corresponding gage pressures being marked on the left-hand side of the diagram.

Reheating Methods.—There are two general methods of reheating: (1) passing the compressed air through a cast-iron chamber or coil of pipe, exposed to a fire or current of hot gases or steam; (2) applying heat within the body of air itself, by the combustion of fuel, the injection of steam or hot water, or by placing in the air main an electric resistance coil.

The first method is usually preferable from a mechanical standpoint, and is reasonably efficient when properly applied. The other method may be employed where the ordinary burning of fuel is not admissible, as in the underground workings of mines.

Design of Reheaters.—Those operating in accordance with the first method, that is, by the external application of heat, are illustrated by the following examples.

The Ingersoll-Rand reheater (Fig. 3) consists of two concentric cast-iron shells, one within the other, the joints being packed with asbestos gaskets, and the inner chamber forming the top of the fire-box. The reheater is a truncated cone, set on a cylindrical fire-box; the cold-air main is connected by a flange coupling at the top and the hot air discharged near the base. Table II gives the dimensions of the two standard sizes and data as to their performance. To reheat more than 550 cu.ft. of free air per minute, two or more heaters are set in series, the air passing from one to another.

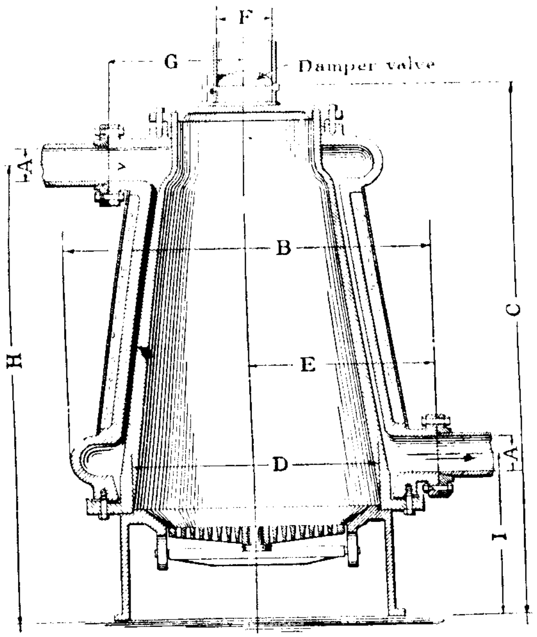

Fig. 3.—Ingersoll-Rand Reheater (Letters refer to Table II).

The Sullivan reheater (Fig. 4) consists of a vertical coil of cast-iron hollow rings, encased in double sheet-steel shells, the space between the latter being filled with asbestos. Below is the combustion chamber, the gases from which pass through the spaces between the air rings. To minimize leakage, the rings are joined by malleable-iron nipples, so that all expand and contract together. These heaters are made in four sizes (Table III).

TABLE II.—INGERSOLL-RAND REHEATERS

Dimensions, in. (Letters refer to Fig. 3)

Nominal size	A	B	C	D	E	F	G	H	I
17	2	17¼	31	10	8⅝	6×4	7½	21⅝	10
41	1	10¼	59 13/16	27½	20½	8×6	15½	51⅞	17¾

Outside diam., in.	Height, in.	Heating surface, sq.ft.	Inlet and outlet, in.	Cu.ft. free air per min., at 80 lb.	Final temp., deg. F.	Approx. increase in vol., per cent	Net weight, lb.
17	31	7½	2	200	250	30	400
41	60	22	4	550	250	30	2,980

TABLE III.—SULLIVAN REHEATERS

Size number	Height, feet	Diameter, inches	Cubic feet free air per minute	Weight, pounds
1	6.5	33	315	2,000
2	7.5	33	575	2,300
3	8.4	33	800	2,750
*	2.1	25 by 20	62–151	275

* Small portable hot-water heater.

A unique design of reheater has been used by the Copper Queen Consolidated Co., Arizona, to heat compressed air for hoisting engines (*Min. and Sci. Press*, June, 1910). A boiler-like tank, 7½ ft. long by 4 ft. diam., with its dome open to the atmosphere, is in a brick setting. The gases from a coke fire on a grate pass through a 10-in. flue in the boiler, heating a residue-oil of 12° Beaumé (flash-point, 700°). The oil, heated to about 335°, is not burned; it surrounds series of 1-in. tubes (making two turns within the boiler), through which the compressed air passes. At the hoisting engine the temperature of the air is 315°.

The inner and outer shells of cast-iron reheaters are subjected to considerable differences of temperature, and the joints between the shells are often difficult to keep tight. In the old Rand reheater (Fig. 5), in which the air passes between the castings in a thin sheet, from the inlet to the discharge, the joint above the fire-box is made by a stuffing ring and packing, to allow for expansion and contraction.

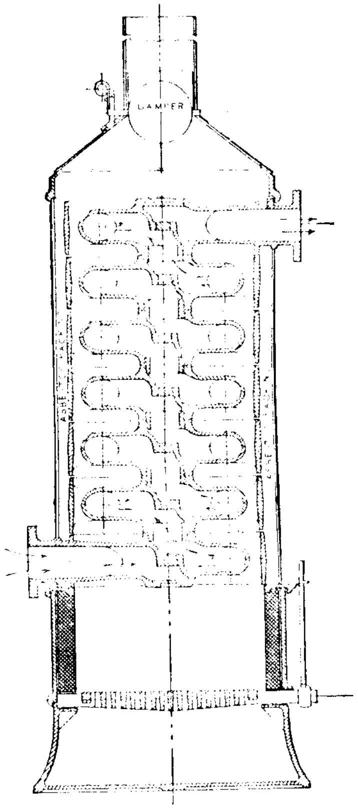

FIG. 4.—Sullivan Reheater.

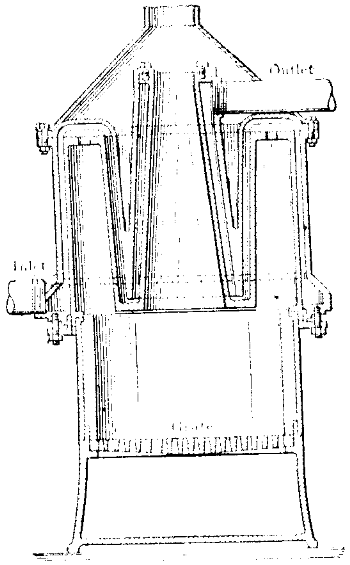

FIG. 5.—Old Rand Reheater.

As the air chamber, in all externally heated or "dry" reheaters, forms in reality a part of the air main, reheating increases the *pressure* only in a small degree. Its real effect is to increase the *volume* of air, which backs up in the main, reducing incidentally the velocity of flow and therefore the loss of pressure due to friction.

Reheaters of the first class (see "Reheating Methods," *ante*) are not economical, because much of the calorific power of the fuel is lost, as is the case with ordinary steam boilers. But the thermodynamic advantage of reheating is so marked that the low efficiency of burning the small quantity of fuel required is of secondary importance.

Reheating by the second method, in which the heat is applied within the body of the compressed air, is especially adapted to use in mine workings, where waste gases from the burning of fuel in the reheater would vitiate the mine atmosphere. For such underground service electric resistance coils, placed in an enlarged section of the air pipe, close to the point of application of the compressed air, have been used successfully, although thus far to a limited extent. They are evidently suitable for stationary air engines, as pumps, winze hoists and small haulage engines. The electric coils are simple and readily installed. A rough modification of this device is to wrap the resistance coils around a short length of the air feed pipe.

The use of superheated steam is also applicable in mine workings. As little as 10 per cent steam obviates the difficulties due to expansion of cold air. At the Coeur d'Alene mines of the Federal Mining & Smelting Co., air at 100 lb. pressure is mixed with superheated steam in the compressor house and piped into the mine for an underground hoist and the rock-drills. This resulted in a substantial saving of boiler fuel for operating the compressor.

With compound air-engines the exhaust from the high-pressure cylinder is sometimes passed through a reheater before going to the low-pressure cylinder. A further benefit may be derived by injecting into the reheater a small quantity of water. The specific heat of water is higher than the specific heat of air; also such part as is converted into steam gives up its latent heat in the air-engine cylinder and prevents trouble from freezing, even with a high rate of expansion. Similarly, it is beneficial to inject a little warm, or even cold, water into the feed-pipe of an air motor.

At the North Star Mine, Grass Valley, Calif., a reheater was placed on the surface near the shaft mouth and the compressed air carried underground by a pipe covered with non-conducting material. Although some saving can thus be realized for moderate distances, it is not practicable for long pipe-lines. In a case on record, where the distance from reheater to air engine was 20 ft., without pipe covering, the gain in power was only 12 per cent, though the absolute temperature of the air at the reheater was increased 38 per cent, with corresponding theoretical increase of volume.

Results of Tests by Prof. J. T. Nicholson, in reheating air from the Taylor Hydraulic Air Compressor, at Magog (Chapter XIV) are as follows: The air was used in a 27-hp. Corliss engine, at a pressure of 53 lb. There were five tests: (1) with cold air; (2) reheating by steam injected into the air main near the engine; (3) the compressed air was passed into a steam boiler, and heated by mixing with the water and steam; (4) the compressed air was blown upon the surface of the water in the boiler, and heated by mixing with the steam; (5) the air was passed through a tubular reheater, fired by coke.

Without reheating, 850 cu.ft. of free air were used per indicated horsepower-hour. By reheating in the boiler, a mixture of 10–15 lb. of steam with the air reduced the air consumption from 850 cu.ft. to 300–500 cu.ft. per indicated horsepower-hour. Thus, one added horsepower was obtained by *wet heating*, by burning 1–1.3 lb. of coal per horsepower-hour. By *dry heating* in the coke-fired reheater, the air was raised to 287° F., and 640 cu.ft. of free air were required per horsepower-hour, or 210 cu.ft. less than with cold air, the saving of air being about 25 per cent. By using 47 lb. coke per hr., 100 hp. in cold compressed air was raised to 133 hp., at an expenditure of 1.42 lb. coke for each added horsepower-hour. Though the reheater used was not very efficient, the fuel consumption was much lower than is attainable in the best boiler and engine practice.

Reheating for Machine Drills has long been a problem. Since the drills in doing their work are generally shifted from place to place at short intervals, the heater must be moved with them to prevent the compressed air from losing its heat and temporary increase of volume.

For *surface excavation* of rock on a large scale fuel-burning reheaters, fired by coal, coke, gasoline or kerosene, are sufficiently portable to follow the drills closely enough to realize some of the benefits of reheating. In such cases, non-conducting covering on the branch air pipes to the individual drills materially increases the efficiency, and should be used even for short distances.

For drilling in *underground mine workings*, reheating is rarely feasible. Fuel-burning heaters can not be employed, for the reasons already stated. Furthermore, the only reheating method of the second class that is at all suitable for underground conditions is the electric resistance coil, and even this is difficult to apply efficiently.

CHAPTER XIX

ROCK-DRILLS

General Classification of Machine Drills.—Although this book deals with compressed-air drills only, the following general classification covering the entire subject of boring holes in rock will be found useful.

I. *Compressed-air drills:*

- *a.* Reciprocating or piston drills, in which the bit is clamped to the piston rod and moves with it (the standard type for more than 50 years, but now obsolescent).
- *b.* Hammer drills, in which the bit is held loosely in the front cylinder head and is struck by rapid, hammer-like blows of a floating piston.
- *c.* Pick machines for coal mining.

II. *Electric drills:*[*]

- *a.* Solenoid type. The bit is attached to a piston, encircled by two solenoid coils placed end to end. A make-and-break current, passing alternately through the coils, causes the piston and bit to reciprocate. The Marvin drill is the only example (invented about 1875).
- *b.* Mechanically-operated drills. In these the bit is reciprocated by a bell-crank and gearing, driven through a flexible shaft by an electric motor.
- *c.* Auger drills, driven through gearing by a very small electric motor mounted on the drill frame (see III, *c*).

III. *Rotary drills:*

- *a.* Diamond and "shot" drills, for deep boring only.
- *b.* Hydraulic drills, in which a toothed bit, under heavy pressure, is slowly rotated through gearing by a small hydraulic engine on the drill mounting. The Brandt drill is the sole example (invented about 1884).
- *c.* Augers, rotated by an electric motor or by hand; for coal and other soft material only. Two unique variations, combining both the auger and percussion principles, are the Ft. Wayne drill, in which an electric-driven auger is struck by rapid blows of floating hammers; and the Pneumelectric drill, in which air is compressed by a small electric motor and utilized in the compressing cylinder.

IV. *Miscellaneous drills:*

Reciprocating type, operated by hand or gasoline motor (unimportant).

Development of Compressed-air Drills.—The reciprocating or percussion drill, invented in a crude form in 1849, was the original type (see historical notes in Chapter I). The first successful design (by Sommeiller) was used in 1861 for driving the Mont Cenis railway tunnel in the western Alps. In the United States, the development of the modern rock-drill dates from the introduction of the Burleigh drill in 1865 at the Hoosac Tunnel in Massachusetts. For mine service, the earliest application of machine drilling on a large scale was probably at the Calumet and Hecla copper mine in Michigan in 1878. Thereafter for many years the reciprocating or piston type of drill was the only one in general use for tunneling and mining.

Beginning about 1890, the principle of the "hammer" drill began to take practical shape, but at least fifteen years elapsed before these drills could be considered serious competitors of the reciprocating machines. All the earlier hammer drills were of small size, capable of making shallow holes only. The invention of the Leyner drill (in 1896), now the Leyner-Ingersoll, marked a new departure. In it the "hammer" principle was successfully applied to machines of a size and capacity, as to diameter and depth of hole, that enabled them to occupy the same field of work as the standard reciprocating drills. Due to their greater efficiency and speed of drilling, hammer drills, of both small and large size, in a variety of designs and a number of distinct types, have since about 1916 almost entirely supplanted the reciprocating drills.

Reciprocating or Piston Drills

Although this type of rock-drill is nearly obsolete, and, with few exceptions, is no longer listed by makers, some are still in service under special conditions; and, as a knowledge of their construction and operation will aid in comprehending the development and mechanical advantages of the hammer drill, several examples will be given. Readers interested in these older drills will find further information in the previous editions of this book.

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[*] A large number of electric drills have been invented and thoroughly tried out in the United States, Europe and South Africa, but with the exception of those operating on the auger principle, none has yet been entirely satisfactory.

General Design and Operation.—Reciprocating drills consist of a cylinder, in which compressed air (or steam) is used, the bit being firmly clamped in a chuck on the end of the piston rod. There are numerous designs of valve motion. In many of them the valve is "air-thrown" by means of auxiliary ports from the cylinder to the chest, which are opened and closed by the reciprocations of the piston; in others the valve is positively moved by a tappet, actuated by the piston.

The rotation of the drill bit on its axis, to keep the hole round and prevent the bit from sticking ("fitchering"), is produced automatically by a rifle-bar, ratchet and pawls. Working speeds are from 300–375 strokes per minute for the larger sizes of drill, to say 450 strokes for the smallest; normal length of stroke in ordinary sizes, 4½–6 in. The admission of air is controlled by a hand valve in the air pipe close to its connection with the valve-chest.

A feed screw, with crank and handle, is carried in a bearing at the rear end of the shell or cradle supporting the drill cylinder, and engages with a nut on the under side of the cylinder casting. The drill head is thus fed forward by hand as the hole is deepened. When the feed has proceeded as far as the length of the feed screw and of the supporting shell permit, the drill is stopped. By reversing the feed the cylinder is moved back, the bit is removed and a longer one clamped in the chuck. The drill head is then fed forward until the new bit touches the bottom of the hole, with the piston nearly at mid-stroke; air is turned on slowly and drilling is resumed. The length of stroke, and therefore the force of the blow, are under the control of the drill-runner. If the drill is fed forward faster than the hole is being deepened the stroke necessarily shortens, because the bit strikes the bottom of the hole before the full length of stroke is reached; conversely, with too slow a feed, the piston will strike the front cylinder head. Thus, the force of the blow may be regulated to suit the conditions. When starting a hole, the stroke should be short until the cutting edge of the bit has adjusted itself to the shape of the bottom of the hole. For hard rock, a short, rapid stroke is best; a longer stroke for soft or tough rock.

Weight of the drill-head and shell (without the mounting) ranges from say 120 to 400 lb., according to the cylinder diameter; a medium size, 2¾-in. cylinder, weighs about 260 lb.

Valve-motions. Variations in the design of reciprocating rock-drills have been mainly in the valve-motions, including the arrangement of the auxiliary or reversing ports. With few exceptions, the

spool-valve has been used, either alone or in combination with an intermediary slide valve.

Nearly all valve-motions fall into three classes: (1) the spool is thrown positively by an arc or a three-arm tappet, which is actuated by contacts with the moving piston; (2) the tappet throws positively a small slide valve, which in turn controls auxiliary ports causing the spool to reciprocate; (3) the spool is "air-thrown" by the piston itself, which opens and closes auxiliary ports from the cylinder to the valve chest.

Some exceptional valve-motions for reciprocating drills, noted in previous editions of this book, are those of the Holman drill, in which two pairs of steel balls, used instead of a tappet, are struck and lifted by the movements of the piston, thus controlling the auxiliary ports to the spool-valve chest; Chicago Pneumatic Tool Co.'s "Gatling" drill, in which a single steel ball, having a very small throw, replaces both tappet and spool-valve (the same design being now used for a hammer drill of these makers); and the Ingersoll-Rand Co.'s "butterfly" valve, now used only in their hammer drills, under which head this unique valve will be discussed. The Temple-Ingersoll "electric-air" valveless drill (no longer made) is interesting in that the reciprocations of the bit are caused by a small electric-driven, two-cylinder, air pulsator, the cylinders of which have hose connections to the ends of the drill cylinder.

The following examples will illustrate the principal features of reciprocating drills:

Sergeant Drill (Ingersoll-Rand Co.).—Fig. 1 is a longitudinal section. The spool-valve and main air and exhaust ports are similar in principle to those of some single-cylinder pumps. Air is admitted on one side of the valve-chest, the exhaust opening being on the other side.

The valve-motion comprises a spool-valve, controlling the main ports, and an arc-shaped tappet, set in a curved slot or seat, between the cylinder and valve chest. The ends of this tappet, projecting slightly into the cylinder, are struck alternately by beveled shoulders of an annular recess in the piston, thus causing the tappet to oscillate at each stroke. In the seat of the tappet are three auxiliary ports, one in the middle and one near each end. These connect with the valve-chest above; the middle port with the middle of the chest, the rear port (nearest the back end of the cylinder) with the *forward end* of the chest, and the forward port with the *rear end* of the chest. In the face of the tappet is a curved slot, just long enough, when in the extreme positions of its throw, to form a communication between the middle auxiliary port and one of the end auxiliary ports. Hence, at each stroke of the piston, the tappet

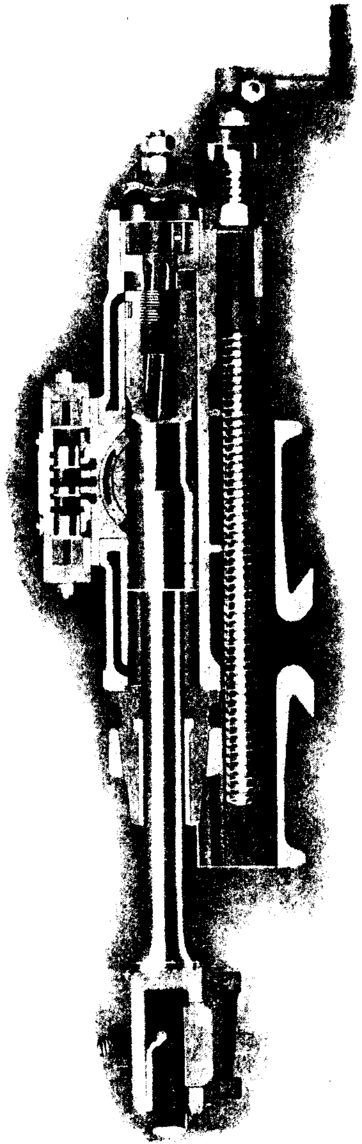

Fig. 1.—Sergeant Piston Drill with Arc Tappet (Ingersoll-Rand Co.).

places the *opposite end* of the valve-chest in communication with the exhaust, thus throwing the spool-valve. In the peripheral surface of the valve is a very fine longitudinal slot, which constantly admits a little live air to both ends of the chest. Thus, when either end of the chest is connected with the exhaust, the valve is thrown toward that end by the air pressure in the other end.

In Fig. 1 the piston is beginning its forward stroke. The valve is admitting air to the back end of the cylinder, the forward end being connected with the exhaust. As the piston advances, the rear shoulder of the recess in the piston strikes the tappet and throws it over. This changes the relation between the auxiliary ports, exhausts the air from the front end of the chest and throws the valve forward, thus preparing for the back stroke of the piston.

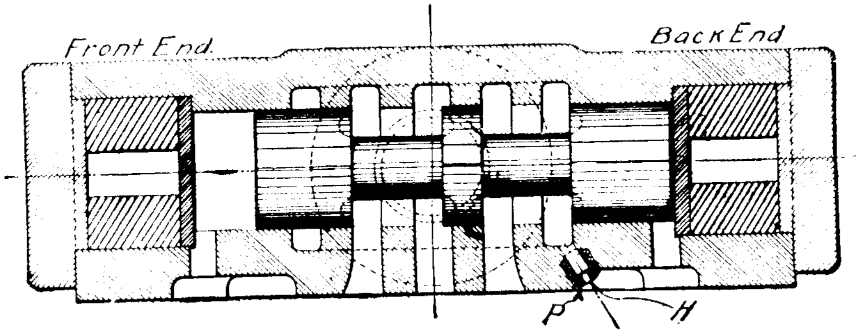

Fig. 2.—Spool-valve and Chest. Sergeant Drill.

The use of the arc-tappet avoids in large part a defect of air-thrown spool-valves, *viz.*, irregularities in operation, due to wear of valve and seat, which permits leakage of air past the valve. Adjustments for irregularities of stroke are made by the device shown in Fig. 2. A brass plug *P*, having a very small hole *H*, admits a little live air to the back end of the chest. Should the piston overrun its stroke and strike the back cylinder head, the area of hole *H* is reduced slightly by peening (hammering) up its edge. This decreases the quantity of air entering the chest, throws the valve a little later, and so increases cushioning in the rear end of the cylinder. If the stroke is too short, *H* is enlarged or cleaned, to admit more air to the chest.

Rotation. A rifled bar, with a ratchet and pawls set in the rear cylinder head, engages with a rifled nut in the hollow piston. The ratchet teeth and pawls (Fig. 3) are so shaped that, on the forward stroke, the piston moves without rotation, the rifle-bar turning the ratchet in its seat. On the back stroke the pawls prevent the ratchet from turning, so that the piston is rotated through part of a revolution. In another design, four instead of two pawls are used. The ratchet ring, with internal teeth, with which the pawls engage, is not fastened rigidly in the back cylinder head, but is held by friction only, under pressure of a cushion spring in the back cylinder head. Hence, if the bit sticks in the hole, and can not rotate freely on the back stroke, the ratchet itself turns, thus preventing injury.

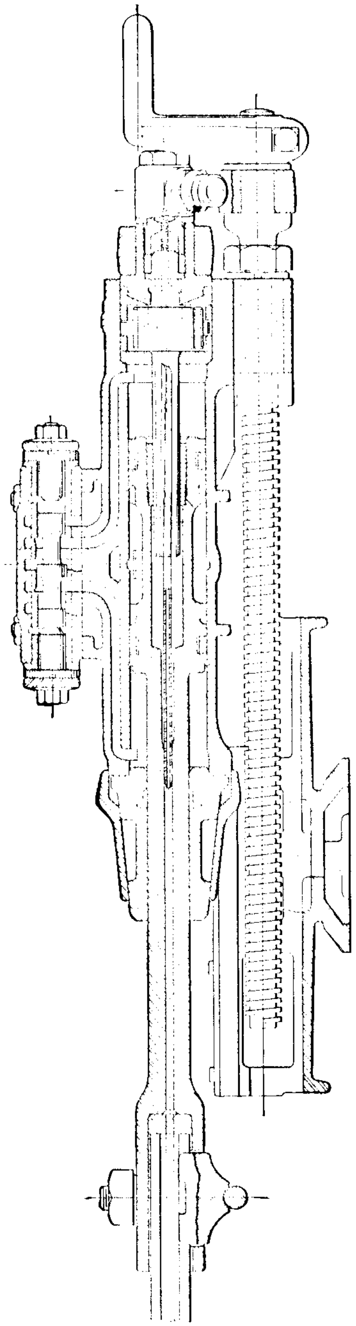

Fig. 4.—"Liteweight" Piston Drill with Air-thrown Spool-valve and Water Attachment. Sullivan Machinery Co.

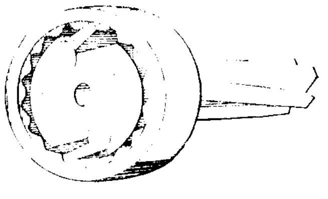

Fig. 3. Sergeant Ratchet and Rifle-Bar.

Sullivan "Liteweight" Drill (Fig. 4) has a three-ringed spool valve, thrown by air admitted from and exhausted directly into the cylinder from the valve-chest, through auxiliary or reverse ports. These ports (not fully shown in the cut) are grooved in the valve-chest, instead of being cored in the cylinder casting. The middle ring of the valve is of slightly larger diameter than the others. As the piston makes its stroke, the reverse ports put the ends of the chest into communication with *opposite* ends of the cylinder, thus throwing the valve and preparing for the next stroke of the piston. The valve will reverse even when the drill head is fed forward until the piston stroke is shortened to only about $\frac{1}{2}$ in. A steel liner, $\frac{1}{32}$ in. thick, is pressed into the cylinder and

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doweled to prevent movement. It gives a hard, smooth surface to take the piston wear, and enables a lighter cylinder casting to be used.

The rotation ratchet has rounded teeth, engaging with two small steel rollers, instead of angular pawls, the rollers being held in place by peripheral springs.

This drill is made both with and without a water injection attachment, for keeping the hole clean. Fig. 5 shows the water type of drill.* The piston, piston-rod and rifle-bar are bored out to permit insertion of a small longitudinal tube, through which water and compressed air pass under pressure into the hollow bit and thence to the bottom of the hole. The water supply is from an 18-gal. tank, connected by hose to the compressed-air pipe and to the drill. A gravity supply of water may replace the tank, provided the water pressure does not exceed the air pressure. The use of the water attachment saves the time ordinarily required to clean out the drill hole.

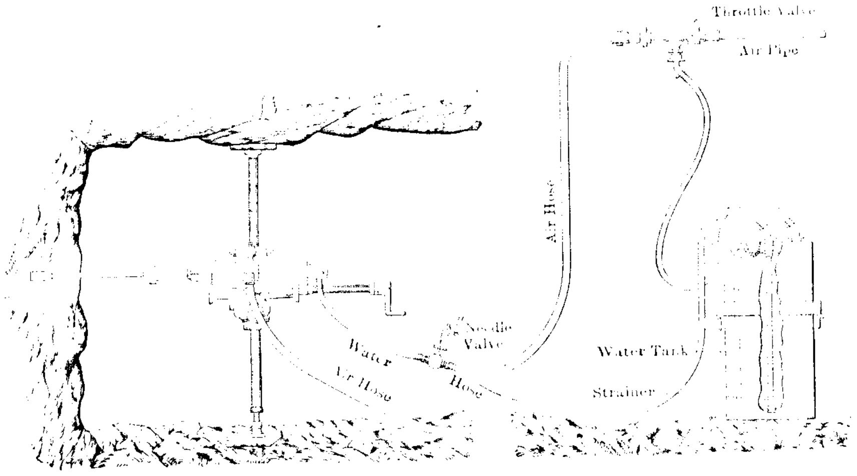

Fig. 5.—Sullivan Piston Drill, with Water Tank and Connections.

Among other reciprocating drills, still occasionally used, are those of the Chicago Pneumatic Tool Co. and Cochise Machine Co., in the United States; and Holman Bros. and Climax Rock Drill & Engineering Works of England. See list of makers, at end of this chapter.

Drill Mountings.—The drill head, comprising the cylinder and its appurtenances and the supporting shell, is mounted on a tripod, column or horizontal bar, or on a carriage. For surface work the tripod only can be used; underground, the tripod, column or bar, depending on the size and shape of the working in which the drilling is to be done. Figs. 6 and 7 show typical mountings for the old reciprocating or piston drills; they are still used under similar working conditions for the large forms of hammer drill, like the Leyner-Ingersoll and the Sullivan (described later).

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* This device is also used for the Leyner-Ingersoll and other large, mounted hammer drills of different makers.

1. *Tripod* (Fig. 6.)—The telescopic legs, hinged by a heavy bolt to the tripod head, can be set to adjust the position of the cylinder and bit for the hole to be drilled. To the tripod head is bolted the shell or cradle, which has guides for supporting the cylinder, as it is fed forward. After the machine is in position for drilling, all bolts are tightened. Weights are usually slipped on the tripod legs, to prevent the drill from shifting while in operation.

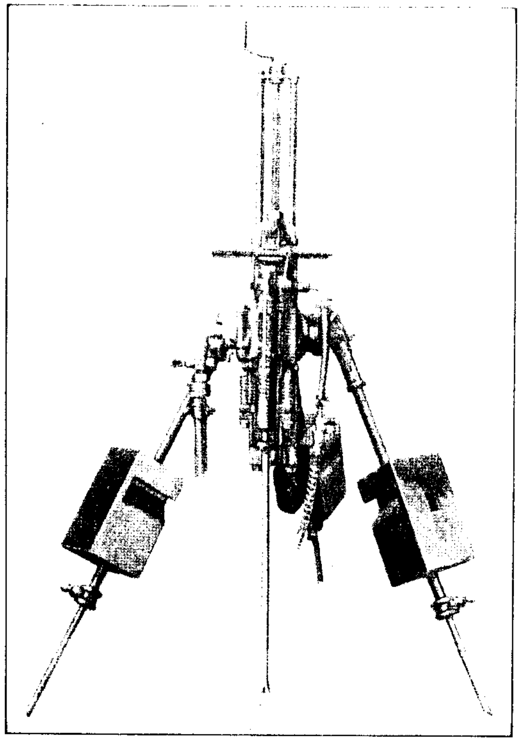

Fig. 6.—Tripod Mounting.

Tripod mountings are required for surface drilling, and also for underground work when the distance between roof and floor, or between the walls, is too great to permit the use of columns. Where a choice exists, the tripod is sometimes preferred to the column, because it can be set up quicker and allows more freedom in locating the holes, for producing the best blasting results. This may be true, also, in sinking shafts of large cross-sectional area, or when the rock is so irregularly fissured that symmetrical rounds of holes are undesirable.

2. *Column or Bar* is a heavy steel tube, 3½ or 4 in. in diameter, according to the size of drill.[*] Being for underground use only, it is set between the roof and floor, or the walls of the working. The lower end of the type of column in Fig. 7 has a cross-piece with two jack-screws. By tightening up the screws, the serrated upper end takes a firm hold

upon the surface against which it is pressed; blocking and wedging are generally required. Another form of column (Fig. 5), with a single, telescopic jack-screw, is useful in small tunnels or mine workings, and, when set horizontally, for shaft-sinking. The drill is carried on an arm, attached to a collar clamped on the column in adjusting the height and angular position of the drill. With the single-screw column the drill is attached directly to the collar, the arm being omitted. Sometimes two drills are mounted on the same column.

The column mounting is convenient for driving mine tunnels, crosscuts and drifts, and the headings of railroad tunnels. It is often used, also, for stoping in veins less than about 7 ft. thick, when the dip of the vein and the method of mining make it difficult to set up tripods.

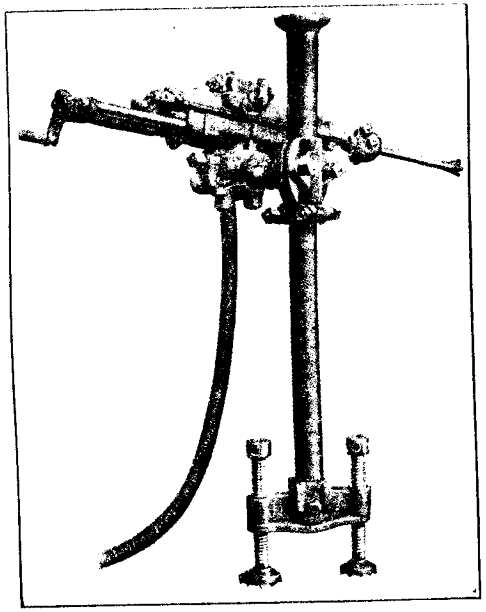

Fig. 7.—Double-screw Column Mounting.

When placed horizontally, as in shaft-sinking, the column is called a "bar." Bars of extra length, for wide shafts, have a pair of adjustable legs hinged at the middle point to carry the weight of the drills without making the bar inconveniently heavy.

For quarry work or similar open-cut excavation, where straight rows of holes are desired, drills may be mounted on "quarry-bars." These are usually 10 to 12 ft. long and 3½ to 5½ in. in diameter, supported at each end by a pair of adjustable legs, resembling those of tripods.

3. *Carriages.*—For driving the headings of large tunnels, from one to four drills may be mounted on a carriage supported on a track. Fig. 8 is a diagram of a carriage for one drill (rarely two), satisfactorily used in several long railway tunnels in the Alps. Fig. 9 shows a standard four-drill carriage recently used in the Cascade tunnel of the Great Northern Railway.

After moving the carriage into position at the face, the horizontal bar, which carries arms for supporting the drills, is tightened by its telescopic jack-screw against the side walls. The drills are then adjusted in position for the desired round of holes. When ready for charging and firing, the carriage is run back on the track, out of danger of injury.

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[*] The drill shown in Fig. 7 is an old-style piston machine.

Fig. 10 shows a specially designed four-drill carriage in the heading of the Moffat tunnel, Colorado, which was completed in 1928. The main horizontal bar carrying the drills is swiveled on a low truck. After the truck has been moved forward to the face of the heading, the bar is set by chain and sprocket gear accurately in position for pointing the holes.

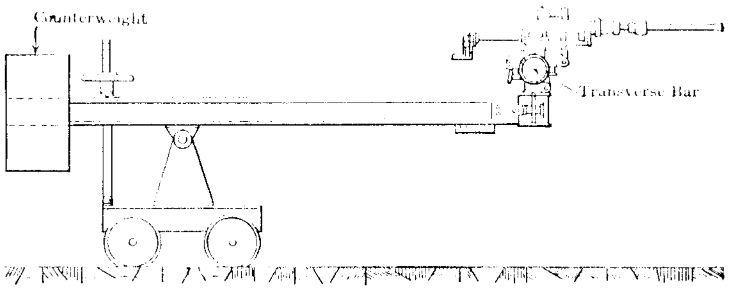

FIG. 8.—Drill Carriage, Loetschberg Tunnel, Switzerland.

For open-cut excavation and quarrying, the wagon mounting (Fig. 11) is convenient for drilling deep holes. The dimensions of the wagon shown in the cut are: length overall, less pole, when moving, 17 ft.; length when drilling, with the guide frame erected in a vertical position, 12 ft.; height overall, when moving, 7 ft. 3 in., when drilling, 18 ft.; width, 5 ft. 9 in.; net weight, 3,000 lb.; length of feed, 15 ft.; depth of hole easily drilled, 25 to 30 ft. A somewhat similar wagon mounting is made by the Ingersoll-Rand Co., for their large drifter drills.

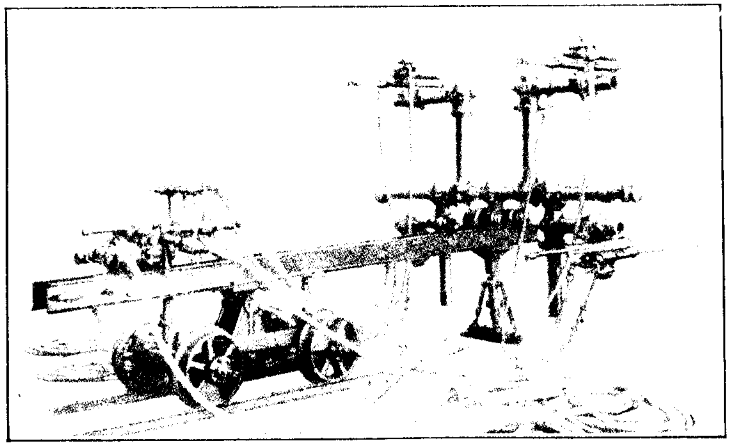

FIG. 9. Drill Carriage for Tunneling. Sullivan Machinery Co.

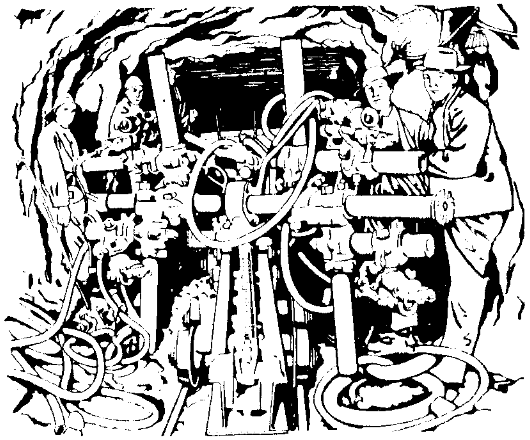

FIG. 10.—Drill Carriage. Moffat Tunnel, Colorado.

HAMMER DRILLS

Since about 1905 the hammer drill has rapidly grown in favor, and is now the standard type for mining and general rock excavation.

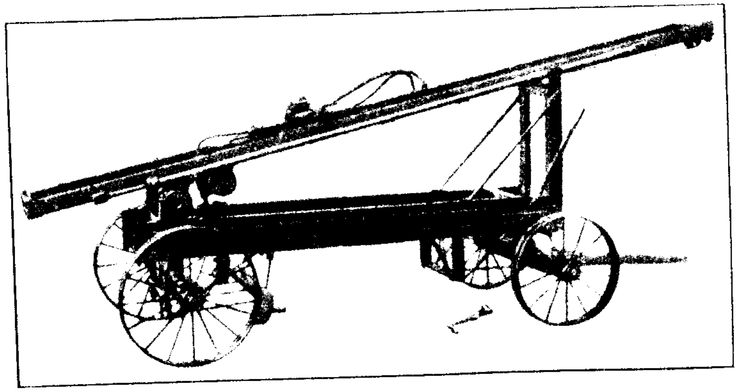

FIG. 11. Long-feed Wagon Mounting for Sullivan Hammer Drills.

Classification, Based upon Structural Characteristics:

A. Machines designed for tripod or column mounting; usually of the larger sizes, and comparable in their field of work to drills of the old reciprocating type.

B. Small hand machines having either a cross- or D-shaped handle;

chiefly for drilling holes directed steeply downward, the weight of the drill resting on the bit itself.

C. Machines having an automatic air-feed standard; primarily for overhead stoping in mines, though they may also be mounted on tripod or column, as for breast work.

Besides the above there is a group of special machines, for various kinds of surface work, known as concrete and pavement breakers, spaders, etc. These differ but little from standard patterns, except in the shape of the bit.

Design and Operation. In the hammer drill, the bit does not reciprocate; its round or octagonal shank, held loosely in the chuck attached to the front cylinder head, projects slightly into the cylinder and is struck a rapid succession of blows by a floating, piston-like hammer. The bit is in constant contact with the bottom of the hole, except during the slight rebound caused by each blow of the hammer.

As the bit does not reciprocate, for holes pointing downward some automatic means must be provided for removing the dust or sludge to keep the hole clean; otherwise the useful effect of the hammer blows is decreased. A hollow bit is therefore generally used. By injecting water, the drillings are displaced and the bit is kept cool. The same result is attained by a jet of compressed air, which produces a low temperature on expanding. In underground work, a mixture of air and water is generally necessary, to allay dust, as well as to clean the hole. Bits are commonly of the four-point cross form. The rosette bit, with six or eight radial cutting edges has some advantages for seamed or other rock difficult to drill. For details of drill bits, see Chapter XXI.

Valves are generally of the spool type. Principal exceptions are: the "butterfly" and "flapper" valves of the Ingersoll-Rand Co., plate ("flap") valve of Holman Bros., and ball valve, formerly used in the "Hummer" drill of the Chicago Pneumatic Tool Co. Several drills of Class B are valveless, the functions of the valve being performed by the movements of the floating piston.

Rotation of the bit, to keep the drill hole round and true, is produced in nearly all drills in two ways: *(a)* by ratchet and rifle-bar, like those of reciprocating drills, but with important modifications (as described later) made necessary by the fact that the bit is not connected to the piston (including all drills of Class A and many of Class B); *(b)* the entire drill is rotated back and forth on its axis by the operator (Class C and some of Class B). An exception to the conventional ratchet and rifle-bar rotation is that of the Gilman drill (Figs. 48*a* and 48*b*), in which the bit is rotated by a small rotary motor

Feed. Class A drills are fed by a hand crank and long screw, like

the old reciprocating machines. Class B, used chiefly for holes pointing steeply downward, are self-feeding, since they stand upon and are supported by the bit. Class C drills have attached to the drill head a long, telescopic standard, which, being supplied with compressed air, keeps the machine fed up to its work as the hole is deepened. Although especially designed for holes directed upward, as in overhead stoping, these drills may also be mounted on a tripod or column for horizontal or breast holes.

When employed for surface work, drills of Classes A and B can be driven by steam instead of compressed air, thereby rendering unnecessary a compressor plant. The use of steam, however, is disadvantageous; it is uncomfortable for the operator, and reduces efficiency and the depth of drilling, because the holes can not be so well cleaned of sludge.

Examples of each class of hammer drill are given on the following pages. It should be noted that, in current drills of Class A, there is a more marked approach to standardization of design in the products of the different makers than was formerly the case. In drills of the other classes, on the contrary, essential variations in details of design are numerous.

CLASS A. MOUNTED HAMMER DRILLS

In the examples of Class A drills given below, together with some of Class B, the present-day approach to standardization, as already noted, is exemplified by the fact that the following three important features of design are identical in principle in all cases, although the minor details are varied by different makers: (1) mechanism for rotating the bit while drilling; (2) use of an axial water- or air-tube and hollow drill steel, for keeping the hole clean; (3) method of feeding the drill. Therefore, to avoid subsequent repetition, descriptions are here given of these parts of the drill mechanism which will apply to all makes of drill unless otherwise specified.

Rotation Mechanism is the same in principle and substantially identical in construction for all drills of Classes A and B, except that of the Gilman drills, as previously noted. For convenience, the letters used in this description refer to Fig. 15. Attached to *a*, which is encircled by a ratchet ring in the back cylinder head is the rifle-bar *b*, which engages with a rifle-nut *c*, screwed into the hollow rear end of the floating hammer or piston *d*. The ratchet, having from two to four pawls and springs, causes the hammer to rotate through a small part of a revolution on each back stroke, the forward stroke being made without rotation. On the front end of the hammer are six or eight straight, longitudinal flutes *c*, engaging with corresponding grooves in the chuck *f*, thus causing the chuck and bit to rotate with the hammer. For a good discussion of the rotation mechanism of drills, see *Eng. and Min. Jour.*, June 9, 1923, p. 1017.

In Fig. 15, the chuck consists of three parts. Variations in chuck design are exhibited in the illustrations of drills of other makers. In the chuck is set the bit, the shank of which has a pair of lugs, or a collar, for locking it in position (see Fig. 16). For bits without lugs on the shank, an "anvil-block" is placed between the hammer and bit; but, as the inertia of this block reduces the effect of the drill blows, it is not generally employed.

Cleaning the Drill Hole is done by a stream of water, or compressed air, or a mixture of the two, transmitted to the bottom of the hole.

In most drills the water or air enters through a valve in the rear cylinder head to a tube of small diameter in the axis of the machine, passing through the hollow rifle-bar and hammer. From the forward end of the tube, the jet is directed into the hollow drill steel, which is in a general use for hammer drills. Fig. 12 shows a typical example. Although this device is automatic, the quantity of water or air admitted is under control of the operator. By thus blowing out the sludge or cuttings, the bottom of the hole is kept constantly clean, thereby increasing the drilling efficiency.

For "wet drilling" water is supplied under pressure from a small tank, as in Fig. 5, which is connected by hose with the compressed-air line. With the water a little compressed air is often used, as the mixture clears the hole more thoroughly than water alone. Holes to a depth of 35 or 40 ft. can thus be kept clean.

In "dry drilling" compressed air only is used.* This serves to depths of about 20 ft. For holes of greater depth, a special "blower" (details of which are given later) is attached to the exhaust port of the drill, and is used intermittently. Full air pressure can be turned on when necessary. This blower is an outside tube, passing into the drill hole alongside of the bit.

Drill Feed. Being mounted on a shell, which supports and guides the drill-head, all machines of Class A are fed forward while at work by a hand crank on a long feed-screw, which engages with a nut under the

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* In dry drilling with the compressed-air jet, too much dust is produced to permit its general use in mines. The dust question has for years received much attention. Although dust from coal, shales and other non-siliceous material is comparatively harmless, that from siliceous rocks or ore is distinctly dangerous, causing "silicosis" (miner's consumption). The enforcement of the use of respirators is always difficult. For a discussion of this important subject, see Peele's "Mining Engineer's Handbook," Second Edition, 1927, pp. 1551 and 1575.

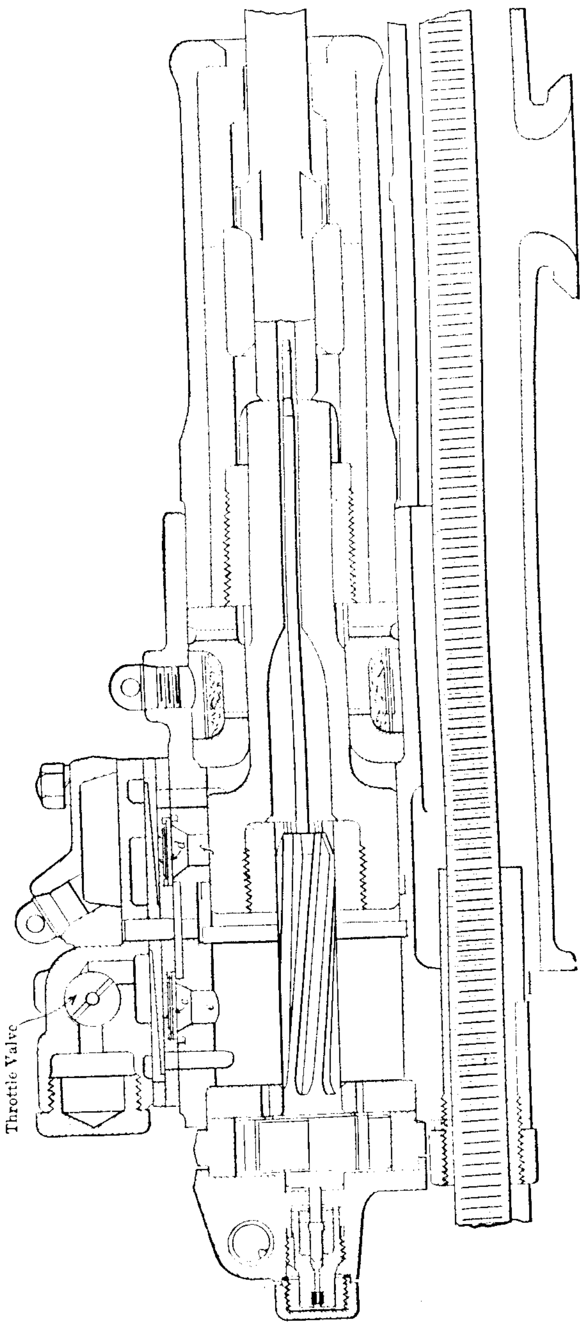


Fig. 12.—Leyner-Ingersoll "Drifter" Drill X-71.

rear end of the cylinder casting. For details, see under "General Design and Operation" of reciprocating drills, the mode of feed being the same.

Leyner-Ingersoll "Drifter" Drills are made in three sizes by the Ingersoll-Rand Co.: N 72, R 72 and X 71, respectively light, medium and heavy weight, differing chiefly in the design of their valves. The X 71 drill (Fig. 12) is a powerful machine, especially suited to deep-hole drilling in open-cut excavation or quarrying that requires holes to say 35 ft. deep. Specifications of these drills, including shell mountings of different lengths, are given in Table I.

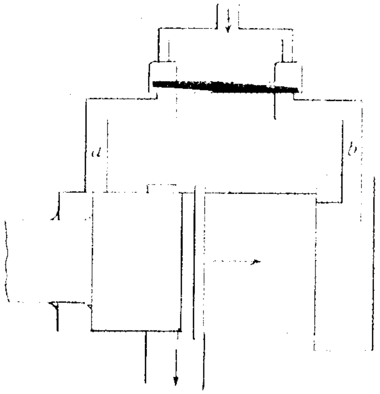


Fig. 13. Valve of the X 72 and X 71 Leyner Ingersoll Drills (Diagrammatic).

In the N 72 and X 71 machines the main valve is of rectangular plate form (Fig. 13), somewhat resembling the "butterfly" valve formerly used in the Ingersoll-Rand "Jackhamers" (see Figs. 40 and 41.) Held in position by a central pin, the rise and fall of the valve's wings control the cylinder admission ports *a* and *b*. The valve is of light weight, with large seating area, and has a cushioned action. Referring to Fig. 12, the oscillations of the main valve are caused by a pair of small auxiliary plate valves *c* and *d*, which have a very slight lift. These are opened and closed by the reciprocations of the hammer, as it passes over the auxiliary ports *e* and *f*, thus alternately admitting and shutting off cylinder pressure acting on the under side of the wings of the main valve.

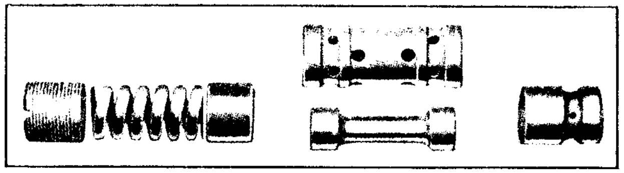


Fig. 14.—Valve Parts, Leyner-Ingersoll Drill R-72.

The valve of the R 72 drill is of the spool type, set in a chest on the side of the cylinder (Fig. 14). It is of simple design, has ample wearing surface, a short stroke, and reciprocates in a replaceable sleeve containing the auxiliary air ports connecting with the cylinder ports. The valve parts are held in position in the chest by a screw plug, which compresses a spring against the sleeve and a buffer in the other end of the chest.

The rotation mechanism of the N 72 drill, comprising the parts shown separately in Fig. 15, is of the conventional type for hammer drills as already described on p. 325, with reference letters to this illustration. In the other two sizes of this drill, the piston and chuck end of the rotation differ in some of the construction details, but are the same in principle.

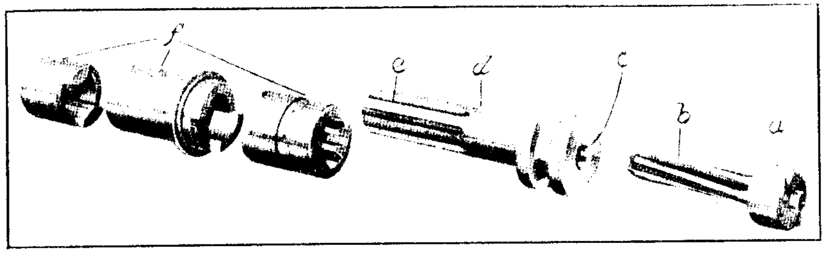


Fig. 15.—Rifle-bar, Piston and Chuck Parts, Leyner-Ingersoll Drill N-72.

Table I. Leyner-Ingersoll Drifter Drills

	N 72	R-72	X 71
Weight, with 24-in. solid feed shell, pounds	125	164	208
Length overall, 24-in. solid feed shell, inches	48⅛	47¾	48¼
Weight, with 30-in. solid feed shell, pounds	137	176	210
Length overall, 30-in. solid feed shell, inches	56¼	53¾	54¼
Weight, with 42-in. solid feed shell, pounds		211	248
Length overall, 42-in. solid feed shell, inches		67¼	68¼
Length of working stroke, inches	3⅛	2⅛	3
Most economical depth of hole, feet	8–10	8–12	10–32

Note.—These drills have 1-in. air hose and ½-in. water hose. They use 1¼-in. round steel, and strike 1,600–1,800 blows per minute.

Bit shanks. For locking the bit in the chuck, the shank has two lugs, for round steel; or a collar, for octagon steel. Fig. 16 shows full details of standard shanks, as used for drills of practically all makes, except that the dimensions may vary.

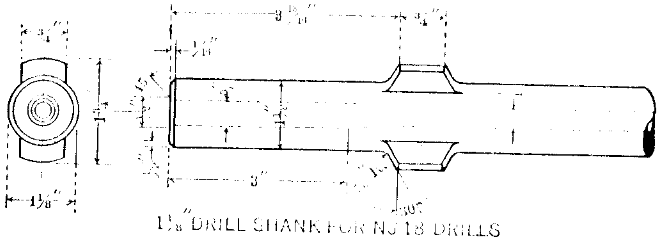
Ingersoll-Rand "Drifter" Drills are made in three sizes, L 74, N 75 and S 70, the head with shell weighing respectively 117, 144 and 185 lb. These drills are alike in design throughout. In many of their features they resemble the Leyner-Ingersoll drifters, differing chiefly in the design of the valve and chuck. Specifications are given in Table II. The L-74 size is a one-man drill. Fig. 17 shows the S-70 machine.

Table II.—Ingersoll Rand Drifter Drills

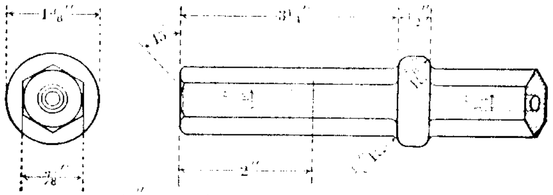
	L 74	N 75	S 70
Weight of drill head, pounds	117	114	185
Length of drill head, overall, inches	$49\frac{1}{2}$	49	56
Height of drill head, overall, inches	$8\frac{3}{8}$	$8\frac{9}{16}$	$9\frac{3}{16}$
Working stroke, inches	$2\frac{11}{16}$	$3\frac{1}{16}$	$3\frac{1}{2}$
Diameter of cylinder bore, inches	3	$3\frac{1}{2}$	4
Air hose, inches	1	1	1
Water hose, inches	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$
Column recommended, inches	3 or $3\frac{1}{2}$	$3\frac{1}{2}$	4 or $4\frac{1}{2}$

Feed shells: L-74, 24 or 36 in.; N-75, 24 or 30 in.; S-70, 30-in. standard, 24 or 42-in. special.

The valve, a thin annular disk called a "flapper" valve, is placed axially in the cylinder in a vertical position, and encircles the shank of the rifle-bar. Its details are shown in Fig. 18. The valve chest, which, with the rotation ratchet, is keyed to the cylinder, consists of two parts containing the auxiliary air ports and holding the valve between them.

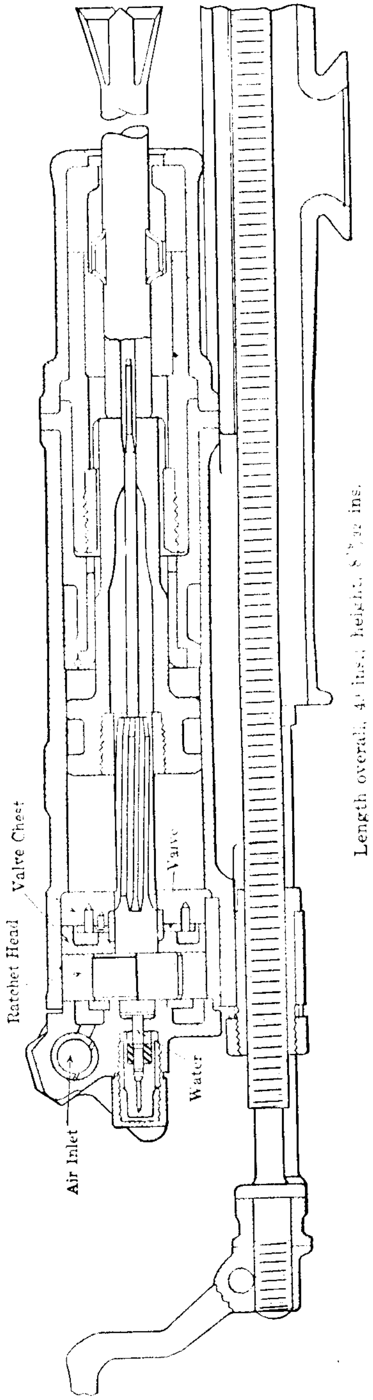


$1\frac{1}{8}$" DRILL SHANK FOR No 18 DRILLS



$\frac{7}{8}$" DRILL SHANK FOR No 26 DRILLS

Fig. 16.—Details of Round and Hexagonal Bit Shanks (Ingersoll Rand Co.)



Length overall, 42 ins.; height, $8\frac{7}{32}$ ins.

Fig. 17.—Ingersoll-Rand Drifter Drill, S-70.

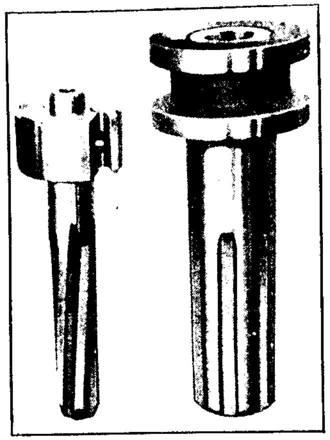


Fig. 19.—Rifle-bar and Piston, Ingersoll-Rand Drifter Drill, S-70.

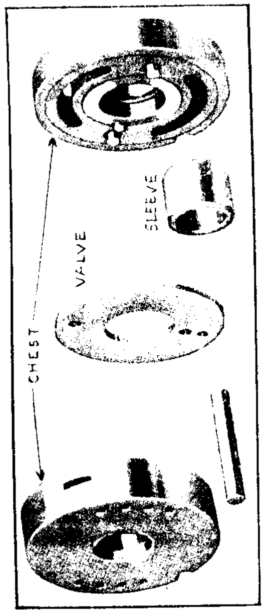


Fig. 18.—"Flapper" Valve and Chest, Ingersoll-Rand Co.

With a small amount of movement, on its central sleeve or thimble, the valve controls short cylinder ports of ample area.

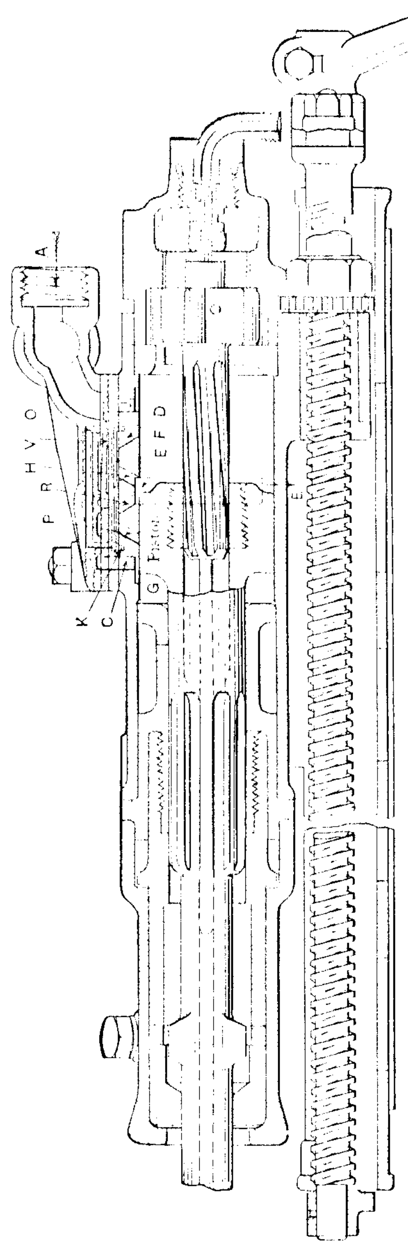


Fig. 20.—Holman Bros.' Drifter Drill PV-160.

Rotation of the bit, in principle as already described, is caused by the "Sergeant" ratchet and rifle-bar (Fig. 19). The ratchet has two pawls of the locked type, so that, when the drill is dismantled for repairs, the pawls with their plungers and springs can not fall out. The chuck is in two parts, the rear member having a collar, which bears against a shoulder in the front cylinder head, thus preventing end-play.

These drills are normally designed for "wet" drilling, a mixture of water and compressed air being used. For "dry" drilling, a "blower valve" and tube are provided, also a special back cylinder head.

Holman Bros., Camborne, England, make the water-feed drifter drill shown by Fig. 20 (see specifications on p. 334).

The unique valve motion comprises a large rectangular plate ("flap") valve, below which are two small relief valves, also of plate form. In the Holman hand sinker drills, PV-31 and PV 57 (described under Class B), the valve design is similar. Referring to Fig. 21, to make the back stroke the compressed air entering at *A* passes over the main valve *V*, through passage *BC*, to the annular space *G* around the forward end of the piston. As soon as the cylinder exhaust port *E* is opened on the back stroke, the air in space *B* above the main valve is exhausted by way of port *C* and space *G*. Then, as the recess in the under side of the valve is charged through port *S* with live air, the valve lifts; and, while thus being held open, the air passing under the valve into the cylinder through port *J* reverses the stroke. The lift of the main valve is less than $\frac{1}{16}$ in., and much less for the relief valves.

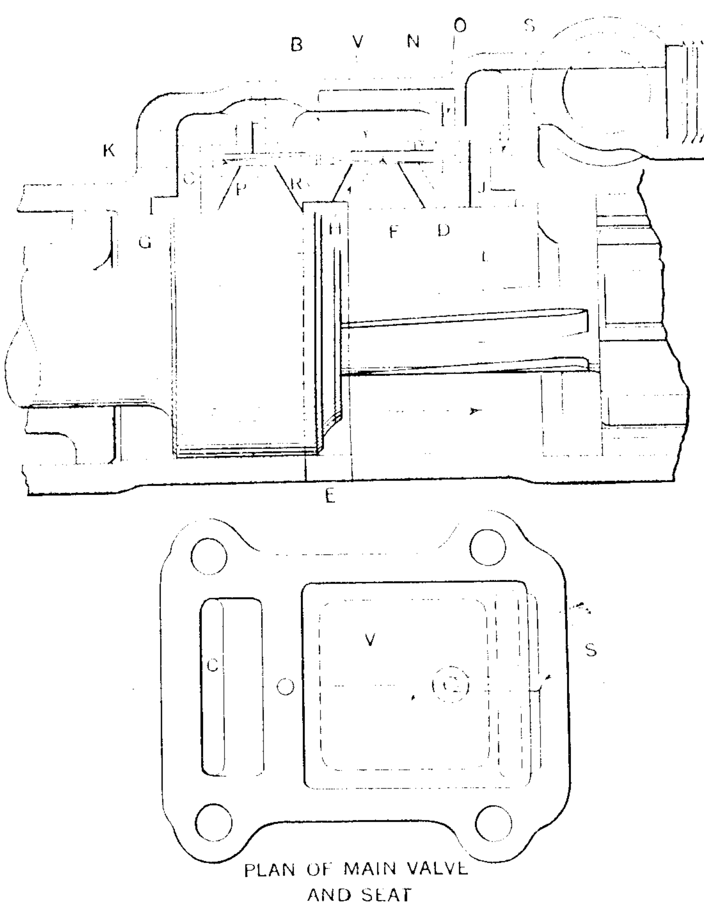


Fig. 21.—Valve Motion, Holman Bros.' PV-160 Drill.

Compression in the cylinder ends is relieved by the relief valves *K* and *F*, as follows: In making the right-hand stroke, the compressed air in space *G*, acting through the auxiliary port *P*, raises relief valve *K*, thus opening communication from port *P* to port *R* and thence, by the annular recess in the cylinder, to the main exhaust *E*. On the left-hand stroke, ports *D* and *H* act similarly with respect to relief *F*. When not in operation, these valves are held closed by live air pressure in ports *O* and *N*.

The details of the rotation mechanism are essentially of the standard type, already described; except that a part of the periphery of the ratchet head is toothed (Fig. 22), the teeth engaging with similar teeth on a lock-nut for the feed-screw nut. The latter is liable to be loosened by vibration. In case the ratchet head rotates slightly, as may be caused by fitchering of the bit, the nut is tightened automatically.

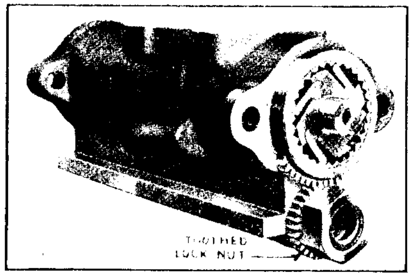


Fig. 22.—Holman Bros.' Rotation Ratchet and Locknut on Feed Screw, for PV 160 Drill.

For attaching the drill-head to tripod or column, the supporting cone is designed to slide on the cradle, and can be clamped at any desired point for setting the machine in position for work.

Specifications: bore, 3¼ in.; stroke, 3 in.; weight, bare, 147½ lb.; weight, with sliding cone, 160 lb.; overall length, 55 in.; length of feed on cradle, 40 in.; additional feed on slide, 28 in.; hollow round drill steel, 1¼ in. The drill can be furnished with anvil block, for octagon or other sections of steel.

Sullivan Water Drill T 3 (Fig. 23), mounted on tripod, column or carriage, is suitable for the same kinds of work as the drifter drills already described. The pressure water is supplied from a small tank, as shown by Fig. 5.

Above and near the rear end of the cylinder is the valve chest, integral with the cylinder casting. The valve, a light-weight, three-ring, differential spool, is air-thrown by the reciprocations of the hammer; large ports communicating with the cylinder admit and exhaust air through the auxiliary ports in the valve chest. In the ends of the chest, which has a replaceable bushing, are single-leaf buffer springs for cushioning the valve action. The ends of the chest are closed by buffers, which, having no threads, are held in place by small side rods.

The design of the rotation mechanism (Fig. 24) is similar in all essential features to that already shown by Fig. 15. A button turned on the forward, fluted end of the piston prevents upsetting of the metal by the blows on the bit, which might cause binding in the chuck bushing. Usually the drill bit has a collared shank for locking in the chuck. When desired, drill steel without shank or collar may be used, an anvil-block (tappet) being then placed between the hammer and the bit.

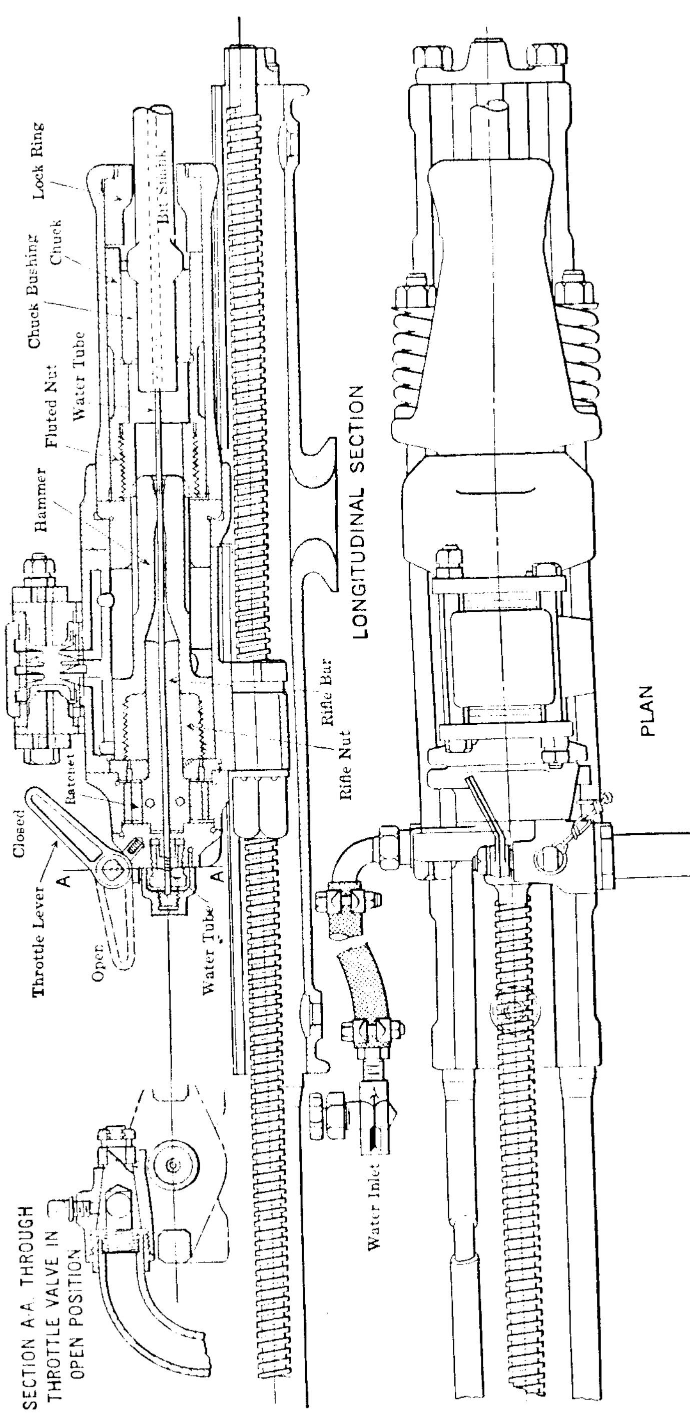


Fig. 23.—Water Drill T-3, Sullivan Machinery Co.

Details of drill: cylinder diam., $3\frac{1}{4}$ in.; net weight, on 30-in. shell, 156 lb.; average net weight of equipment, 30 lb.; shipping weight, 216 lb.; air hose, 1 in.; water hose, $\frac{1}{2}$ in.; drill steel, $1\frac{1}{4}$ in.; maximum diameter of hole drilled, $2\frac{1}{4}$ in.; depth of hole easily drilled, 12 to 18 ft., maximum depth, 40 ft. The drill may be fitted with 24-, 30- or 36-in. shell.

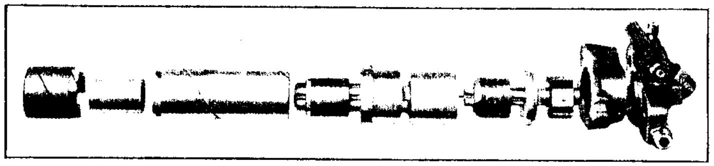


Fig. 24.—Rotation mechanism, Sullivan T-3 Drill. Parts, from right to left back head, ratchet ring, ratchet and rifle-bar, ratchet-collar, rifle-nut, piston, buffer ring, retaining bushing, chuck, chuck bushing, and lock-ring for bit.

Chicago Pneumatic Tool Co. makes two "drifter" drills: type CP-5, serving for average drilling conditions, and CP-6, especially designed for deeper holes in hard rock or where seams, etc., may cause difficulty in rotating the bit. For specifications of both machines, see Table III.

The general features and rotation mechanism of these drills are of the conventional form, as already described. Fig. 25 shows the CP-6 drill and Fig. 26 its valve action.

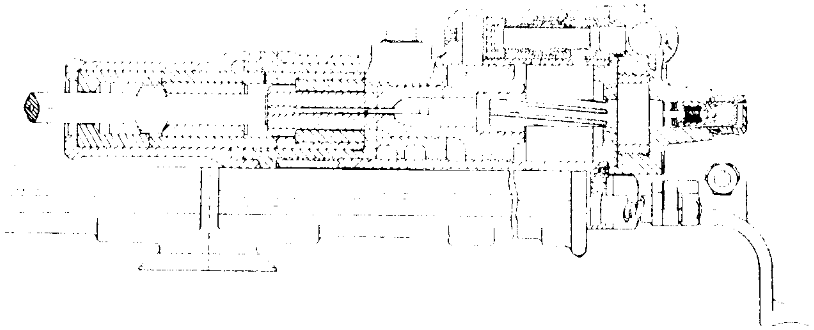


Fig. 25. Drifter Drill CP-6. Chicago Pneumatic Tool Co.

Referring to the upper diagram of Fig. 26, live air enters at *1*; passing thence through the hollow spool valve *2*, auxiliary ports *3* and the main cylinder port *4*, to the front piston chamber *5*, it causes the back stroke of the piston. During the first part of this stroke, the main exhaust *23* and auxiliary port *6* will be closed; then, when port *14* is also covered by the piston, pressure is built up against the valve shoulder *15*, *16* by air entering from the interior of the valve through the small rear diagonal holes *17*. This tends to offset the pressure on the opposing

valve shoulder *19*, *20*, which has been similarly built up (after the valve "holding port" *18* has been covered) by air admitted by the forward

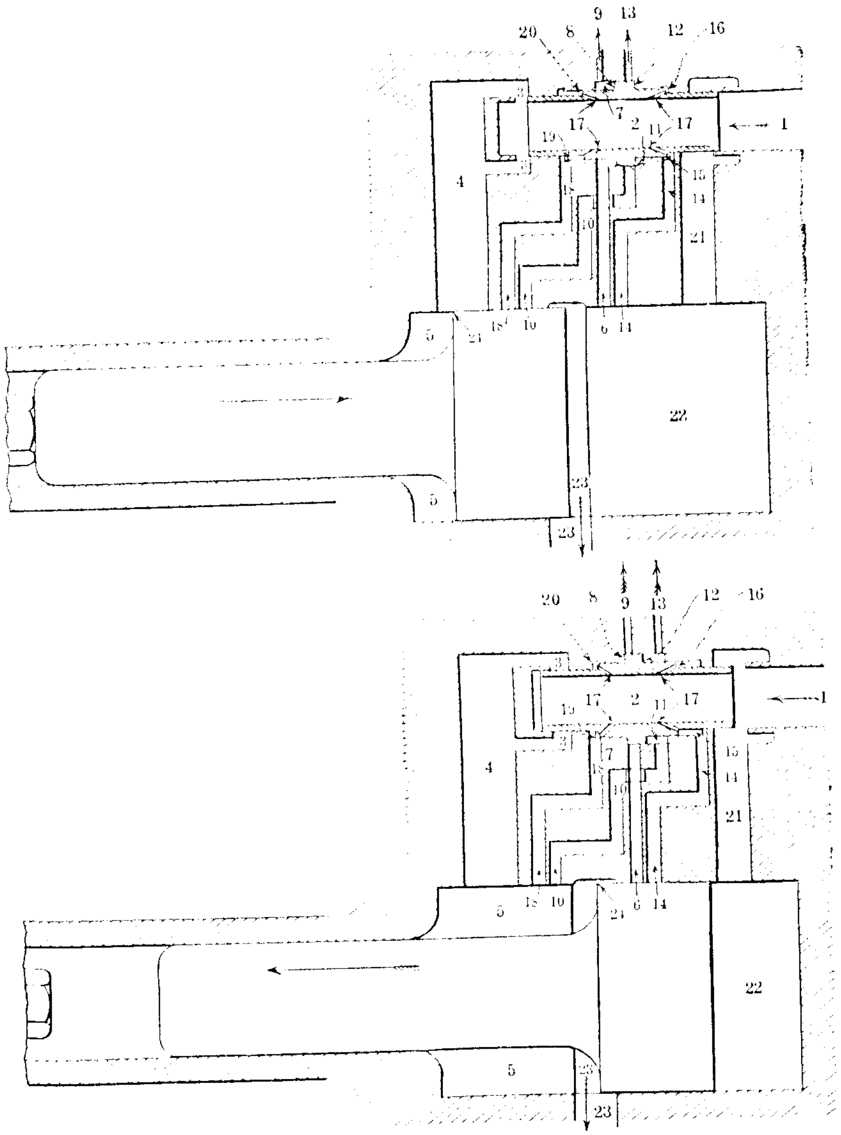


Fig. 26.—Valve Action, CP-6 Drifter. Chicago Pneumatic Tool Co. (Diagrammatic.)

diagonal holes *17*, so that the valve for the time being is held positively in position.

As the piston continues rearward, its front edge *24* uncovers port *18*, thus somewhat decreasing the pressure in the chambers and valve shoulder at *19*, *20*, when the pressure in piston chamber *5* falls sufficiently. Meantime, no live air can enter at "trip port" *10*, which is covered by the piston, while the trip-valve face *12* is open to the atmosphere through *13* and trip-valve face *8*, with its chamber *7*, are also open through *9*, *6*, *22* and *23*.

Moving still farther rearward, the piston finally uncovers port *10*, which is large as compared to port *13*. Live air from piston chamber *5* then acts on valve shoulder *11*, *12*; and, since the pressure on the forward valve shoulder is about balanced (as stated above), the valve moves forward to the position shown by the lower diagram. In this position, the valve admits live air through *21* to chamber *22* and drives the piston forward, thus completing the cycle.

The valve action of drifter CP 5 differs materially from the above. It is identical with that of sinker drill CP 8, in connection with which it will be described.

Fig. 25 shows the standard design for "wet" drilling. For "dry" drilling, as in sinking, the water connection can be replaced by a special air valve, admitting live air to the axial tube.

TABLE III.—CHICAGO PNEUMATIC TOOL CO.'S DRIFTER DRILLS

Size	CP 5	CP 6
Diameter of cylinder, inches	3	$3\frac{1}{2}$
Length of stroke, inches	$2\frac{1}{4}$	
Length of feed, inches	24	
Length overall, inches	46	
Diameter of air hose, inches	1	
Diameter of water hose, inches	$\frac{1}{2}$	
Standard round hollow drill steel, inches	$1\frac{1}{4}$*	
Size of lug shank, inches	$1\frac{1}{4} \times 3\frac{13}{16}$	
Suitable tripod, pounds	287	486
Diameter of suitable column, inches	3 or $3\frac{1}{2}$	$3\frac{1}{2}$
Net weight of drill head and shell, pounds	120	145
Economical drilling depth, feet	10 14	10 20

* On special order, chucks are furnished for 1 or $1\frac{1}{4}$-in. round steel with lug shank; or hexagon or quarter-octagon steel, in both the anvil block and collared-shank types.

Gardner-Denver Co. makes the following drifter drills; Nos. 7 and 17, weighing respectively 124 to 141 lb. and 178 to 185 lb., according to the length of the guide shell; also Nos. 7711 and 107 (Fig. 27), recent lighter-weight models. Their designs have been developed from those of the well-known Waugh drills, the patents of which are now owned by the Gardner-Denver Co. The Waugh "Turbro" drifter is also made by this company. Drill No. 7, when unmounted and provided with a spring handle (weighing then only 100 lb.), can be used for sinking.

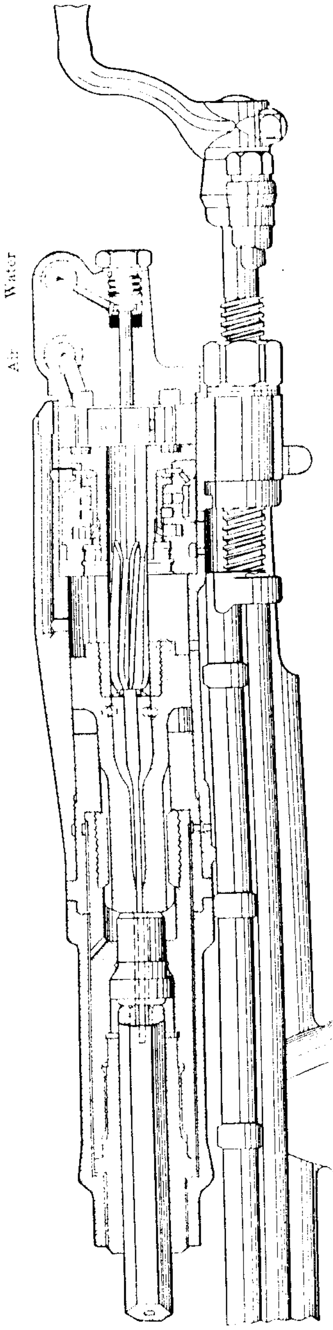


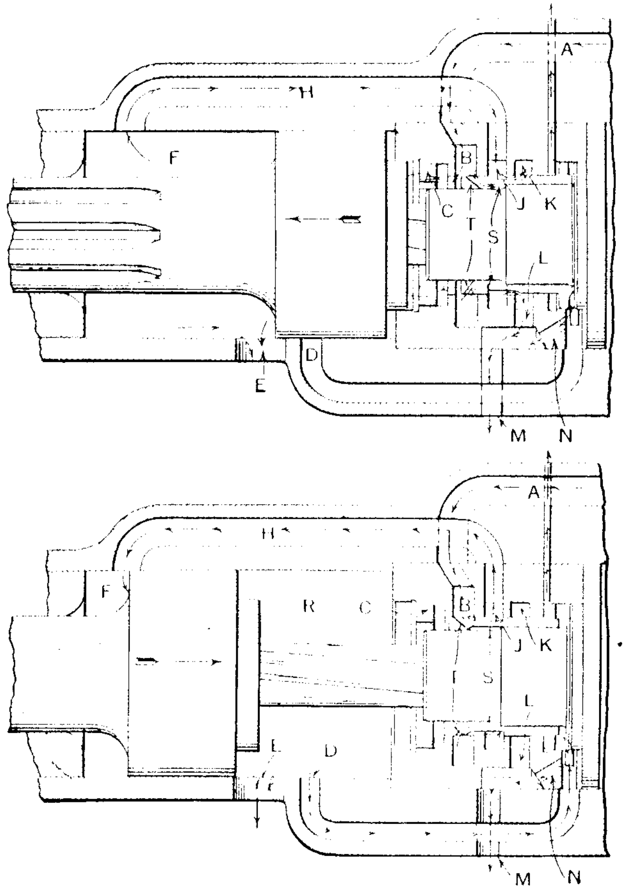
FIG. 27.—Gardner-Denver (Waugh) Drifter, Model 107.

The rotation mechanism of Nos. 7, 17, 7711 and 107 is of the usual ratchet and rifle-bar type. For the "Turbro" drifter, rotation is caused by a small air turbine attached to the cylinder casting (for details see description of Model 37, under Class B drills).

The valve, tubular in form, is contained in a housing in the rear end of the cylinder, and encircles the rifle-bar shank. Figs. 28 and 29 show the valve and hammer at the ends of their back and forward throw.

When the hammer is in its rear position (Fig. 28), live air entering port *A* from the throttle passes to the annular recess *B*, and thence through a series of holes *C* in the valve chest to the rear face of the hammer. This causes the forward stroke, the air in cylinder space *F* being meanwhile released through the main exhaust *E*. As the hammer uncovers "trip port" *D*, air from behind the hammer passes to an annular chamber at the rear of the valve. Since the area of the valve's rear face exceeds that of its front face, the valve then moves forward, thus cutting off admission of air through *B* to the rear of the hammer, and opening passage to the front. Advancing still farther, by its inertia, the hammer delivers its blow. At this instant, the front end *F* of the cylinder, though no longer in communication with the main exhaust, is still open to exhaust through port *H* into groove *J* encircling the valve, and thence by groove *K* and ports *L* and *M* to the atmosphere. This additional exhaust from the front end prevents cushioning of the forward stroke.

While the hammer is at the end of its forward stroke (Fig. 29), the rear end *R* of the cylinder is open to exhaust at *E*, and also through trip port *D* to the chamber at the rear of the valve and out by ports *N* and *M* to the atmosphere. The back stroke is then caused by live air passing from *B* into valve groove *J*, and thence through port *H* to *F*. The valve remains in its forward position until the hammer on its back stroke covers trip port *D*. In this position of the valve, exhaust groove *K* is closed, and compression is built up in the rear cylinder chamber *R*. The



Figs. 28 and 29.—Valve Action, Gardner Denver (Waugh) Drifters.

inner surface of the valve has two diameters, the resulting differential shoulder forming pocket *S*, which is always connected by the diagonal holes *T* in the valve with live-air groove *B*. The pressure in *S* holds the valve in its forward position, and prevents fluttering, until this pressure is overcome by the greater total pressure against the front end of the valve. When compression in *R* becomes sufficient to overcome the holding pressure inside the valve, the latter moves to its rear position, completing the cycle. The small vertical port above the rear end of the valve relieves cushion pressure.

Cleveland Rock Drill Co. makes four sizes of drifter drills: D5, D7, D8 and D9, models D7 and D9 having been brought out in 1929. Fig. 30 shows their general construction. All have the same valve action and are similar in design. Model D5 is now approaching obsolescence. For specifications, see Table IV.

The valve is a hollow spool, encircling the rifle-bar shank. Its action is shown diagrammatically by Figs. 31 and 32. When the piston is at the end of its forward stroke, the valve is also in its forward position, admitting air to the front end of the cylinder, and vice versa. Entering throttle *A*, the air passes from *B* through holes *C*, between the ratchet-ring and rifle-bar; then by port *D* to the annular distribution chamber *E*, which is constantly under live air pressure.

With the piston in its rear position (Fig. 31), air passing through the

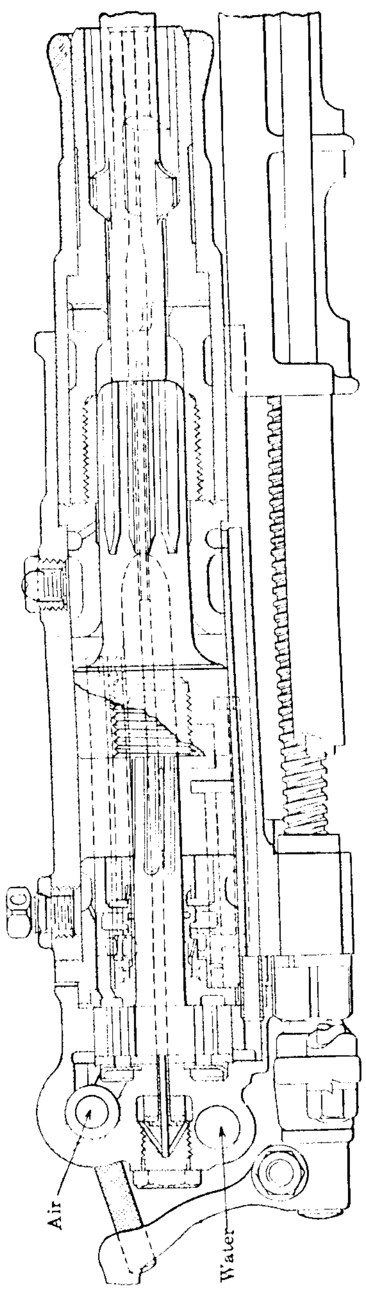
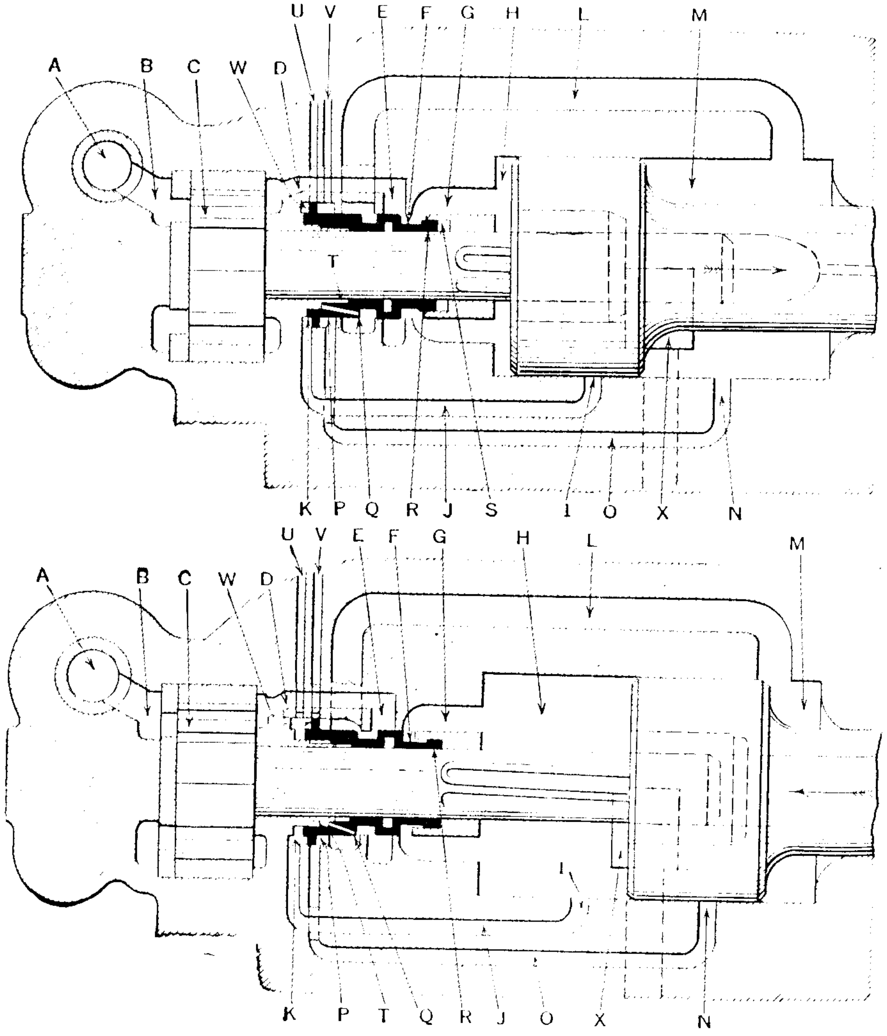


Fig. 30.—Cleveland Rock Drill Co., D-8 Drifter Drill.

annular valve recess F and port G to the cylinder acts on the rear end H of the piston, causing the forward stroke. When the rear edge of the piston passes "trip port" I, air flows through J to K, thus throwing the valve forward. As the piston advances, it uncovers the main exhaust port X, discharging the air from the cylinder.



Figs. 31 and 32.—Valve Action of the Cleveland Drifter Drills, the Valve being shown at the Ends of its Throw.

With the piston in its forward position (Fig. 32), live air in E flows through port L into the front end M of the cylinder, causing the back stroke. As its front edge passes trip port X, air flows through O into the annular recess P, throwing the valve to the rear, into its position in Fig. 31. Then, when the piston again uncovers port X, exhaust takes place, completing the cycle.

To prevent fluttering and irregularity of action, the valve is held in its respective back and forward positions by the "holding holes" R and Q, which admit live air to the small recesses S and T. The "vent holes" U and V are important in exhausting the "trip" air from each side of valve flange W, after the valve has shifted. Without these vents, the valve action would be sluggish.

Table IV.—Specifications of Cleveland Drifters

Model	Cylinder diam., in.	Stroke, in.	Length over-all, in.	Available feed, in.	Weight, with 30-in. shell, lb.	Weight without shell, lb.	Economical drilling depth, ft.
D7	$3\frac{1}{8}$	$2\frac{3}{8}$	$46\frac{1}{2}$, with 30-in. feed	30–41	115	72	8–12
D8	$3\frac{3}{8}$	$2\frac{1}{4}$			144	100	10–18
D9	$3\frac{5}{8}$	$2\frac{3}{4}$			146	102	12–25

Climax Rock Drill and Engineering Works, Carn Brea, England, make two sizes of drifter drill (Fig. 33), and the "Joburg" No. 3, which is suitable for similar work (see Table V). Most of the sinker and stoper drills of this company (noted later), can also be mounted on column or tripod, for drifting.

Table V.—Specifications of Climax Drifter Drills

Model	Cylinder diam., in.	Stroke, in.	Weight with cradle, lb.	Length overall, in.	Blows per min.	Air consumption at 80 lb., cu.ft.
Light drifter......	3	$2\frac{1}{4}$	134, 28-in. feed 140, 40-in. feed	51	1,500	125
Heavy drifter.....	4	$2\frac{3}{4}$	198, with 40-in. feed	51	1,450	150
"Joburg" No. III.	$2\frac{3}{4}$	$2\frac{9}{16}$	192	$48\frac{1}{2}$	950	110

Fig. 34 is a diagram of the valve action of the drifter drills (a different type of spool-valve action is used in the "Joburg" design). The valve chest is set transversely on the rear end of the cylinder; not as shown for convenience in the diagram. As the central ring of the spool valve has a slightly larger diameter than the ends, the entering live air holds the valve in position until acted upon by the opening and closing of the ports. Passing into the valve's interior through the small diagonal holes *H*, and following the arrows, the air acts upon the outer ends of the valve, to throw it when either of the auxiliary ports *VR* and *VF* is covered by the piston.

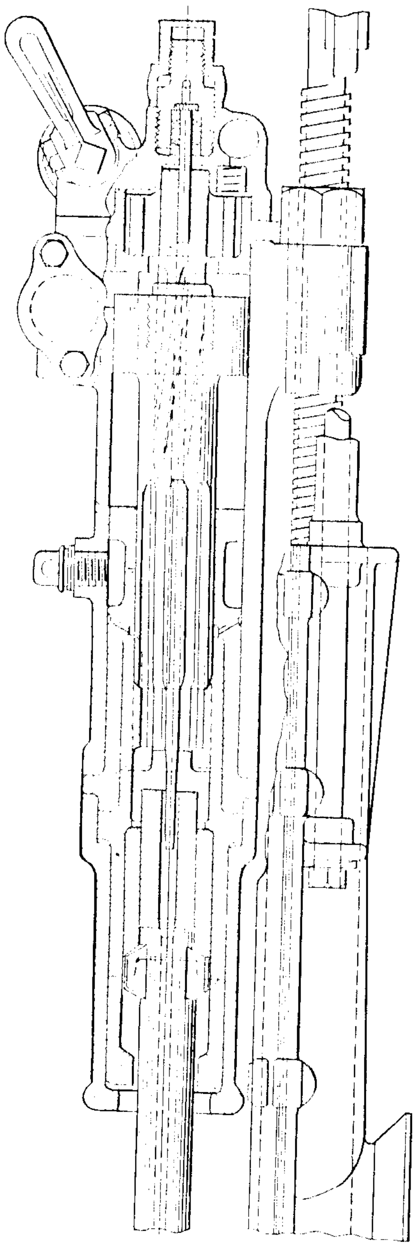


FIG. 33.—Climax Rock Drill & Engineering Works. Light Drifter.

With the piston at the end of its back stroke and the valve at the end of its forward throw, as in the diagram, air enters by main port *RP* into the rear of the cylinder, causing the forward stroke. In the first part of this stroke, auxiliary port *VR* is closed, and, since port *VF* is open, the front end of the valve chest is in communication with the main exhaust *E*. Meantime, the air in front of the piston is also exhausting through *E*. As the piston nears the end of its stroke, auxiliary port *VF* is covered and *VR*, being then open, puts the rear end of the valve chest in communication with the exhaust. This reverses the valve, which admits live air through the main port *FP* to the front of the cylinder, and the back stroke ensues.

To reduce cylinder compression (cushioning of the piston stroke), at each end of the valve chest there is a circular groove *AR* in the valve bushing, which is open to the atmosphere through the release ports indicated by dotted circles. By this means, a little air can still flow out of the cylinder ends at completion of the stroke, after the main exhaust *E* has been covered by the piston.

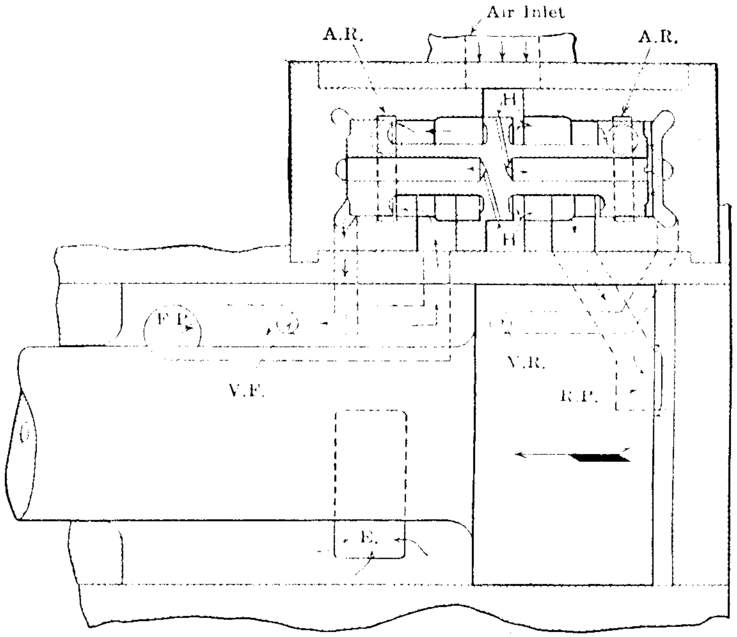


FIG. 34. Valve Action of the Climax Light Drifter (diagrammatic).

Gilman Manufacturing Co. makes drifter drills differing in some of their features from the foregoing examples, but chiefly in the design of the rotation mechanism. Instead of the conven-

TABLE VI. GILMAN DRIFTER DRILLS

	Number	Cylinder		Weight, lb.
		Diam., in.	Stroke, in.	
Screw-feed	86	2½	2	88
	87	2½	2½	89
	88	2⅝	2½	100
	89	2⅝	3	103
	421	2¾	2⅝	115
	25	3	3	125
	26	3	3½	126
	50	3½	3	150
	51	3½	3½	151
	321	3¾	2¾	175
Air-feed	19	2½	2½	116
	20	2⅝	2½	120
	21	2⅝	3	123
	521	2¾	2⅝	120
	69	3	3	169
	80	3½	3	191
	AX521	3¾	2¾	230

tional rifle-bar and ratchet, the bit is rotated by a special motor, independently of the hammer's movements. For details of the rotation see under Class B drills, Figs. 48*a* and 48*b*, with the accompanying description.

These drills have "horse-shoe" plate valves, and use the air expansively. The smaller sizes are suitable for general mine service; the larger, for heavy tunnel work. Sizes of cylinder and net weights of the screw- and air-feed types are given in Table VI.

Some of the drifters are designed with interchangeable features, rendering them applicable for a variety of duty. For example, the cylinder, hammer and valve plate of the 3½-in. by 3-in. screw-feed drill (No. 50) can be varied to secure a faster and lighter or a slower and heavier blow. The standard drill of this size, working with 100 lb. air pressure, delivers a 100 ft.-lb. blow, 2,200 times per minute.

CLASS B. SMALL HAMMER DRILLS, WITH HANDLE

Drills of this class comprise: (*a*) those which are rotated by the operator; (*b*) those provided with automatic rotation mechanism, closely resembling that of Class A machines. All are "one-man" drills.

Primarily, as their name implies, these light-weight machines are intended to be held in position by the operator, the weight of the drill resting on the bit. They are especially designed for holes pointing steeply downward, as in shaft-sinking, underhand stoping and similar work. For many years this was their only field of service. Most makers, however, now supply special cradles, for mounting on tripod or column such drills of this class as have automatic rotation, thus adapting them to drilling holes in almost any direction.

When changing bits while drilling a hole, it may be found that a strong pull is required to loosen the bit for withdrawal. Hand hammer drills are therefore provided with a "steel puller," consisting of a yoke on the forward end of the drill, through which the steel passes. On raising the drill sharply, when the yoke comes into contact with the collar on the shank of the bit, the steel is pulled out of the hole.

The following examples exhibit the principal features of Class B drills. Certain special tools, like concrete breakers, clay spaders, chipping hammers, etc., are omitted, as they are outside of the scope of this book. See remarks near top of p. 321.

As nearly all the numerous drills of this important class have many points in common, full statements regarding the several machines of each maker would involve much repetition. Therefore, in the examples, the illustrations and descriptions are limited to such matters as will aid in comprehending the outstanding variations in design.

Hardsocg Wonder Drill Co.'s valveless, hand-rotated drill (Fig. 35) is made in five sizes, weighing 12, 20, 30, 31 and 45 lb., all using hollow steel. Air consumption is from 12 to 36 cu. ft. free air per minute at 80 lb. The larger sizes are employed only for down holes. The hammer in its reciprocations acts as a valve, and is the only moving part.*

Air is admitted at nipple 2 (to which are attached the throttle and hose) and, entering annular recess 3, acts constantly on shoulder 13 of the hammer. On beginning the forward stroke, ports 5 and 6, through the head of the hammer, are opposite recess 3, and admit live air into and behind the hollow hammer. Since the area thus presented to air pressure is much greater than the area of shoulder 13, the hammer is driven forward. Just before the hammer strikes the bit, ports 5, 6 reach the annular recess 8, through which part of the air that has caused the forward stroke goes to exhaust port 4, and part through the hollow bit to the bottom of the hole. The exhaust having taken place, the back stroke is made by the constant pressure of the inlet air on annular shoulder 13 of the hammer.

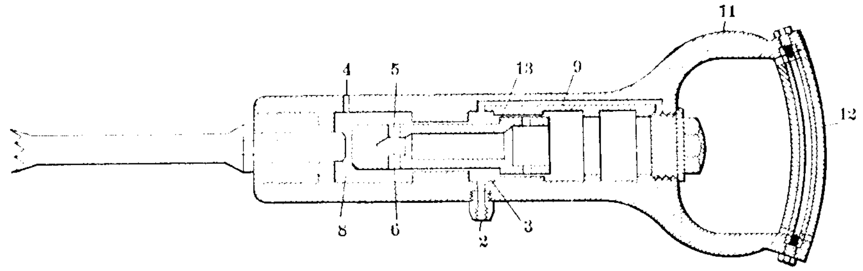


FIG. 35. Hardsocg No. 1 Drill, with D-Handle.

This drill can be converted into a stoping or drifting machine by removing the large plug and the handle 12, and screwing into the rear cylinder head an air-feed standard, like that of Class C machines. For making breast holes, the modified drill is mounted on a light column.

Fig. 36 shows the Hardsocg drill, with automatic rotation: *a* is the ratchet box, with spring-controlled pawls; *b* the rifle-bar, engaging the rifle-nut *c* in the rear end of the hollow hammer *d*, which is thus rotated like the piston of a reciprocating drill. The forward end *e* of the hammer was formerly of square cross-section, reciprocating in the bushing *f*, which held the octagon shank *g* of the bit and rotated it. The end *c* is now fluted, with corresponding grooves in the bit bushing. A yoke-shaped retainer *h* holds the bit in its socket. This drill weighs 46 lb. and uses from 31 to 36 cu. ft. free air per minute at 80 lb.

---

* This drill was one of the earliest of the hammer drills.

Live air enters port *i*, from the throttle *j* to the annular recess *k*, and, as shown by the arrows, passes alongside the rifle-bar, through passages *l,l*, in the rifle-nut, to the rear of the hammer. On completing the forward stroke, the air in the rear end of the cylinder passes to the exhaust port *m*. When the hammer is in this position, the small shoulder *n* is opposite the annular recess *k*, thus admitting air to act on the large shoulder *o* of the hammer, and causing the back stroke. At the end of the forward stroke, some of the exhaust air passes down alongside the square head of the hammer (which makes a loose fit in bushing *f*), and goes through the hollow steel for cleaning the drill hole. If required, by turning the throttle *j* backward, live air can also be delivered through the bit, by way of port *p*.

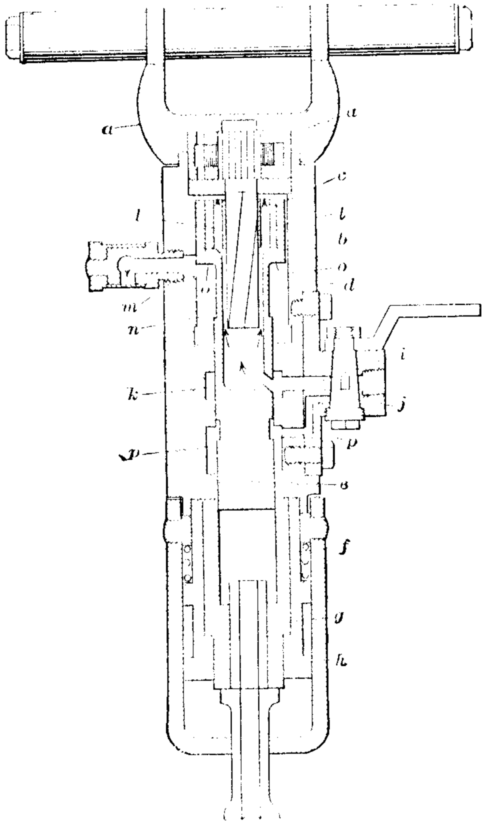


FIG. 36.—Hardsoeg No. 7 Drill, with Automatic Rotation.

The Hardsoeg Co. also makes the following drills: Nos. 2 and 5, having spool valves, but otherwise resembling Fig. 36; Nos. 5-W and S-W, which are light-weight "water" drills, respectively spool-valve and valveless, and are provided with special shells for tripod or column mounting.

Ingersoll-Rand Co.'s "Jackhamer" Drills are made in nine sizes or patterns, specifications of which are given in Table VII. The design of the first six is based upon the patents of J. Geo. Leyner (1896–1909), acquired by the Ingersoll-Rand Co. in 1912, the original drill of the series being the BCR-430 (Fig. 39).

TABLE VII.—SPECIFICATIONS OF INGERSOLL-RAND "JACKHAMER" DRILLS

	BAR 33	BBR 13	BCR-430	R-72	DCR-23	DDR-13
Diameter of cylinder, inches	2⅛	2⅜	2¼	2⅝	2⅝	2⅝
Net length, without steel holder, inches.						
Dry	15½	18	21	20¼	22½	24½
Wet	......	19	21	20¼	22½	24½
Auger	......	18	....	17¼		
Weight, with connections, pounds:						
Dry	27½	40	47½	49	57	75½
Wet	......	45	51	52	61	78
Auger	......	39½	....	42		
Size hexagon steel, inches		⅞		Dry, 1 Wet, 1 Auger, ⅞		1
Steel shank, inches		⅞ × 3¼		1 × 4¼ ⅞ × 3¼		1 × 4¼
Recommended depth of hole, feet	3-6	8-10	10-12	10-14	12-16	12-25

Diameter of air hose for all sizes, ¾ in. Lengths of drills, including steel holder, are 1½ to 2¼ in greater than the net lengths in the table.

Besides the above drills, the Company in 1928 added the following new "Jackhamers":

Number	Type	Cylinder diam., in.	Stroke, in.	Length, in.	Weight, lb.	Steel	Air hose, in.
R-39	Dry Blower Wet	2⅝	2	20¼ 20⅞ 20⅞	43 43 46	⅞ or 1 in. hex.	¾
S-49	Dry Blower Wet	2⅝	2½	23¼ 23⅜ 23⅜	57 57 60		
X-59	Blower Wet	3	2$\frac{11}{16}$	24½	72 75	Round or 1 in. hex.	1

Sizes S-49 and X-59 are especially suited to deep hole drilling.

These lines of drills include machines suitable for every kind of mine work and open-cut excavation. They are similar in general mechanical design, but differ in their valves and some of the details of construction. The smaller sizes have "Butterfly" valves; the others, in the first table, spool valves. The new drills, listed below Table VII, are fitted with "flapper" valves (see Figs. 17 and 18, under Class A drills and the accompanying description). Their throttle valves have three positions, instead of two: with the handle "up," air is shut off; in the intermediate position, the drilling power is light, accompanied by heavy "blowing"; when the handle is "down," full drilling power is obtained. The exterior lines of those three machines also differ from the older models, the only projecting part being the exhaust blower valve.

Except the light BAR-33 drill, intended especially for shallow holes, as for block-holing, pop shots, trimming, etc., "Jackhamers" are made in both "dry" and "wet" patterns; that is, for cleaning the hole, air only or both water and air can be used. In addition to having the usual axial tube, as in drifter drills, some sizes, designed for deep holes or work in heavy, sticky ground, are equipped with an auxiliary "hole-blowing" device (described below), to insure thorough removal of the sludge. The BBR 13 and DCR 23 drills, when thus modified, are called respectively BBRA 13 and DCRA 23. Special blowers are used for other sizes also.

All drills of the series have the conventional automatic rotation on the back stroke. Except the BAR 33, all may be mounted on special shells (cradles), for attaching to tripod or column, thus fitting them for drilling holes approaching the horizontal, as in tunneling and drifting.

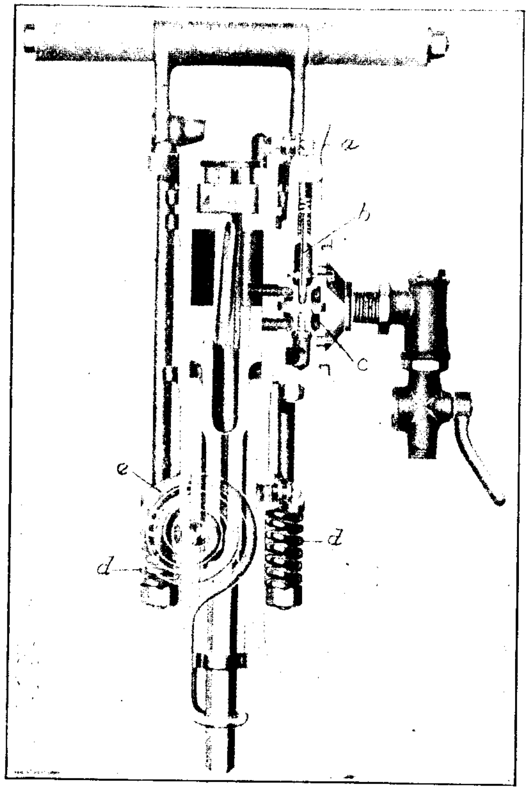


Fig. 37.—Ingersoll-Rand "Jackhamer" Drill DCR 23.

The DCR 23 machine (Fig. 37) will serve to illustrate the construction of the heavier "Jackhamers." It is suitable for deep-hole work in shaft-sinking and road and quarry excavation. The valve is a spool, in a chest cast integral with the cylinder. To furnish ample blowing capacity, for keeping the hole clean, this drill has a special blowing device. By pressing inward the latch *a*, on top of the valve chest and

just under the handle, a pin *b* is pushed against the valve *c*, holding it in its forward position and thus forcing the piston backward. This permits passage of full-pressure air through the bit to the bottom of the hole. The same device is used in the DDR 13 drill.

Operation of the drill is eased by the heavy side springs *dd*. The bit is held firmly in the chuck by pressure of the double helical spring *e*. The handle has rubber grips. This drill, and all others of the series, are lubricated by the "heart-beat" lubricator (described later).

The design of the new series of drills (see second part of Table VII) is shown by Fig. 38.

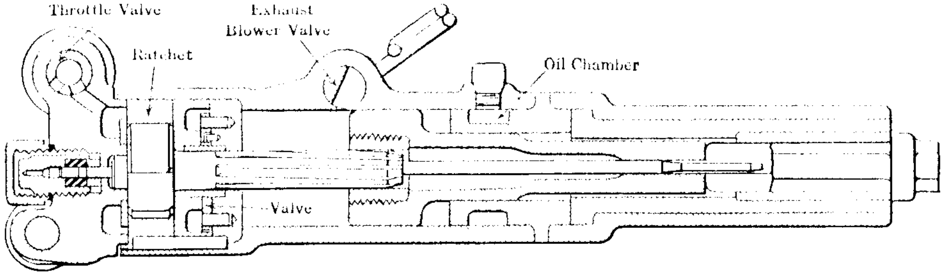


Fig. 38.—Ingersoll-Rand "Jackhamer" S-49.

Fig. 39 shows the BCR-430 "Jackhamer." For "dry" drilling a special valve is screwed into the exhaust end of the valve chest, so that the operator can close the machine's exhaust while the inlet throttle is open. A strong flow of live air thus passes through the hollow steel to the drill hole. For "wet" drilling a swiveled water connection and strainer are attached to the rear end of the cylinder, water being supplied under pressure from a 6-gal. tank connected by hose to the air pipe, in the manner shown by Fig. 5. The water pressure should be not less than 25–30 lb., but must always be less than the air pressure.

The "Butterfly" valve, used in this drill, is shown diagrammatically by Figs. 40 and 41. It consists of two flat wings with ground seating surfaces, with a central trunnion. The valve oscillates in a slot in the chest, being actuated by the unbalancing of air pressure alternately on the wings. A very small travel gives a large port opening. There are four ports, opening into the faces of the valve slot, a separate inlet and exhaust for each end of the cylinder. The inlet ports open opposite to each other at one end of the slot, the exhaust ports at the other end. Hence, when one valve wing closes the inlet to one end of the cylinder, the opposite face of the other wing closes the exhaust port from the other end. One inlet and one exhaust port are therefore always open.

Fig. 40 shows the beginning of the forward stroke. The rear inlet port S_2 and the forward exhaust port E_1 are open, the others being closed. Since the piston now covers rear exhaust port E_2, the pressure of the inlet air holds the valve to its seat over port S_1. When the advancing piston uncovers E_2, live air passes through the cylinder to E_2, almost balancing the pressure on the valve wings. But the resultant of the forces acting on the valve, comprising the flow of exhaust air in E_1, the impact of inlet air on entering the chest, and the friction of the air in passing through S_2 and E_2, keeps the valve on its seat, in its first position, until near the end of the forward stroke. When the piston closes the exhaust port E_1, the cushion pressure produced in front of the piston acts through port S_1, and throws the valve to the position in Fig. 41. The back stroke now begins, live air entering through S_1, while the exhaust escapes through E_2. This completes the cycle.

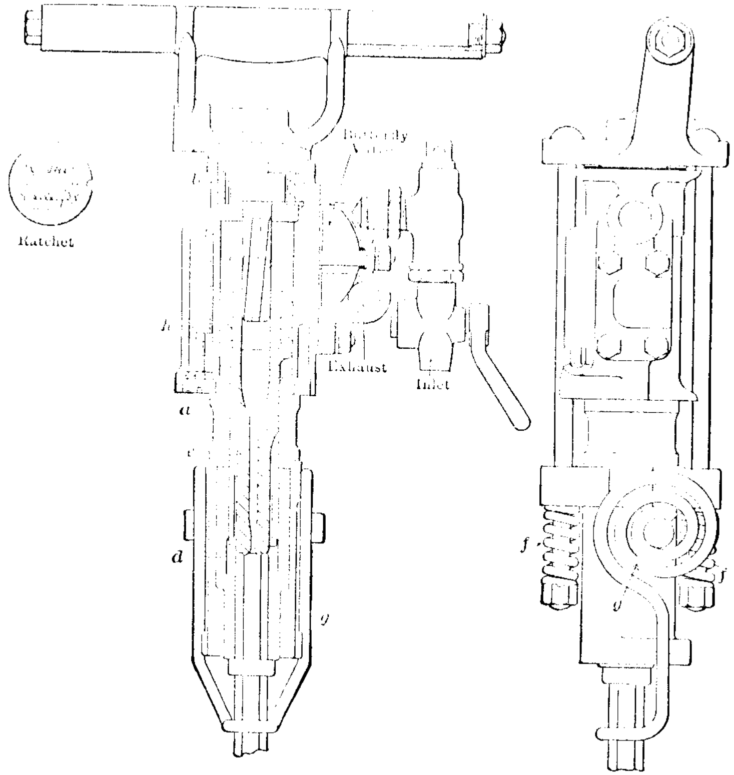
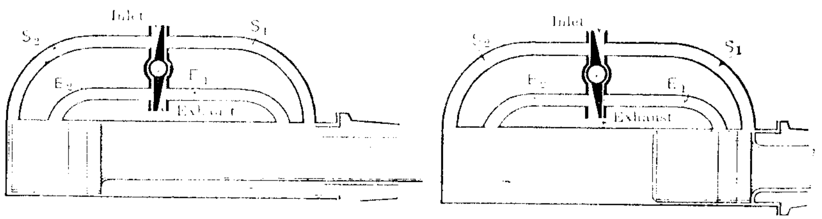


Fig. 39. Ingersoll-Rand "Jackhamer" Drill BCR 430.



Figs. 40 and 41. Ingersoll-Rand "Butterfly" Valve (diagrammatic).

Other "Jackhamers" having the "Butterfly" valve are the BBR 13 and DCR 23.

For extra heavy down-hole work, the N 72 Sinker is recommended. It weighs 104 lb. and is essentially the Leyner-Ingersoll N-72 Drifter, except that a handle replaces the feed shell and a positive blowing mechanism is provided resembling that of the dry BCR 430 drill. It is $26\frac{1}{2}$ in. long and uses $1\frac{1}{4}$-in. steel.

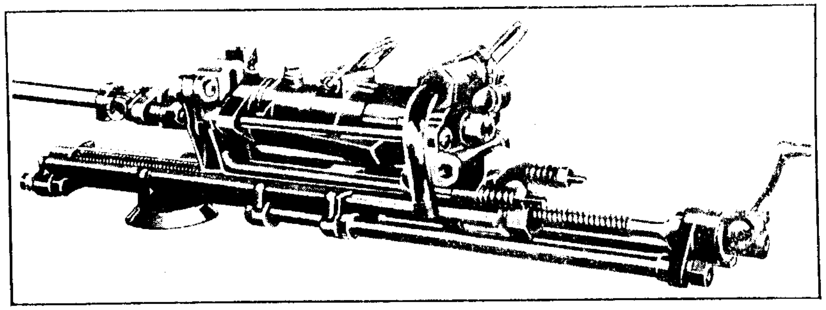


Fig. 42.—Ingersoll-Rand "Jackhamer" S 49 on Cradle Mounting.

For drilling flat holes, as in drifting or breast-stoping, the S 49 Jackhamer may be mounted as shown by Fig. 42. The cradle, resembling that of an ordinary drill, is varied slightly in design according to whether it is to be used for the dry or the wet machine; it is adapted to either column or tripod. A supplementary cradle, carrying the drill head, rides on the feed screw of the main cradle.

For drilling with auger or spiral steel having a cruciform bit (Fig. 43), as is common in mining coal and soft iron ore, the BBR 13 "Jackhamer" is provided with *down-stroke rotation*. The bit is thus revolved on each forward stroke, which aids in removing the cuttings. With this modification the drill is known as the BBR 230.

Ingersoll-Rand "Imperial" Drills (types MV 1 and MV 2) are valveless (Fig. 44). They resemble the Hardsocg drill, already described, in that the piston (or hammer) acts as its own valve. The hollow piston *a* has an enlarged part *b* at the rear, against the shoulder of which the air pressure acts constantly, and 6 slot-shaped ports *c*. In the cut the piston has completed its forward stroke; air is being exhausted through the piston ports *c* to the exhaust port *d*. The return stroke is caused by the constant air pressure on the shoulder of *b*.

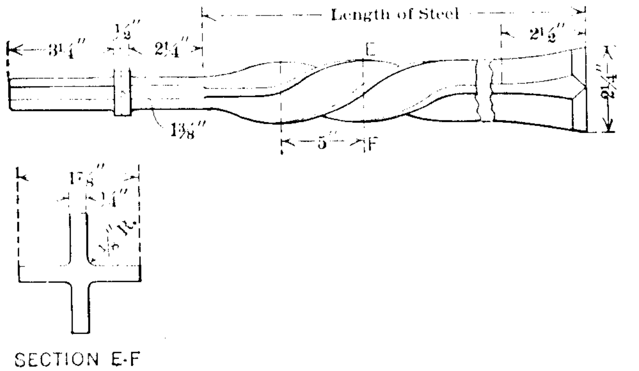


Fig. 43. Twisted Cruciform Bit, for Drilling in Coal.

The drill is rotated by a handle attached at *c*. The bit shank is held in bushing *f*, made to receive hexagonal or cruciform steel. Solid steel is used for these machines, which are designed for shallow "plug holes." Weight of drill, 42 lb.

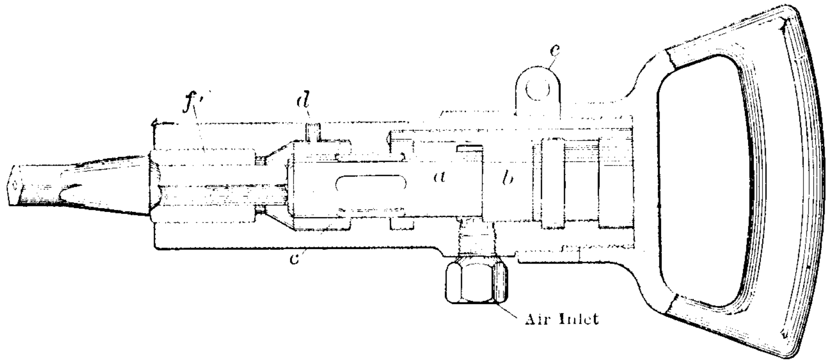


Fig. 44.—Ingersoll-Rand "Imperial" Hammer Drill (Types MV-1, MV-2).

Sullivan Machinery Co. makes a line of one-man drills, as specified in Table VIII. In addition to the drills listed in the table are the following: a light "plugger," DH-3 (weight, 20 lb.), a heavy water-tube "rotator" sinker, L-3 (weight, 100 lbs.), and an auger machine, H-8 (weight, 39 lb.), for drilling in coal, hematite, or soft, broken rock.

The last named has down-stroke rotation, similar to that noted in connection with Fig. 43.

Table VIII.—Sullivan "Rotator" Drills

	L-7, air tube	L-7, water tube	L 7	L 8	L 8	DP 321	DP 321, light model	DP-331, steam-driven
			Solid or hollow piston		Water tube			
Cylinder diameter, inches	2¼	2¼	2¼	2¼	2¼	2¼	2¼	2¼
Overall length, inches	23¼	23¼	23¼	20⅛	20⅛	17⅞	17⅞	24
Weight, pounds	47	47	45	39	40	35	31	51
Maximum diameter of hole, inches	2⅜	2⅜	2⅜	2	2	2⅜	2⅜	2⅜
Normal drilling depth, feet*	12	12	6–12	12	12	12	12	6

* Under favorable drilling conditions, greater depths can be reached.—Diameter of air or steam hose, ¾ in. Hexagon shank, for all the above drills, ⅞ × 3¼ in. If shankless steel is used, a tappet block is inserted in the chuck, between the piston and end of the drill steel.

As the name "Rotator" implies, all the Sullivan drills except the DH-3 have automatic rotation, which is of the conventional form already described, resembling in its details that of the drifter drill, Fig. 23. When desired, cradles are supplied for the L 7 and L 8 machines, so that they can be mounted on tripod or column, for drilling approximately horizontal holes, as in drifting or breast stoping.

Fig. 45 illustrates the general features of the "rotator" drills. Each has a light-weight, three-ringed spool valve, in a chest cast integral with the cylinder. In the ends of the chest are single-leaf springs, for cushioning the valve's movements. The air and water connections are shown respectively at *a* and *b*. Water passes through a screen to the rear end of the axial tube, and is mixed with exhaust air. When extra strong blowing pressure is required to clean the hole, the button *c* is pushed upward by the operator. This forces the valve to the rear of the chest and holds the piston in its rear position, thus admitting live air through port *d* to the forward end of the cylinder and thence into the hollow bit.

By the air pulsation of each stroke a small quantity of oil is discharged from the oil chamber *e*, through the ball-valve *f*, to the ratchet and then through holes in the ratchet collar into the cylinder. The chuck parts are oiled through the plug *g* in the side of the cylinder.

Fig. 46 shows the steam drill, DP-331. Steam enters by the by-pass throttle *a*, through the pipe *b*, to the axial tube. The operator can admit steam simultaneously to the valve chest and pipe *b*, or to the latter only. Thus, if necessary, the hole can be cleaned finally after the drill has been stopped and the bit withdrawn. The drill handle is covered by a non-conducting canvas jacket.

Although the use of steam has the advantage of making a compressor plant unnecessary, steam-driven machines can drill holes only about one-half as deep as by air, as has already been stated, and they are applicable only to surface work. For the usual intermittent operation of rock drills, about $1\frac{1}{2}$ boiler hp. is required per drill, this estimate being based upon a steam pressure of 70 lb.

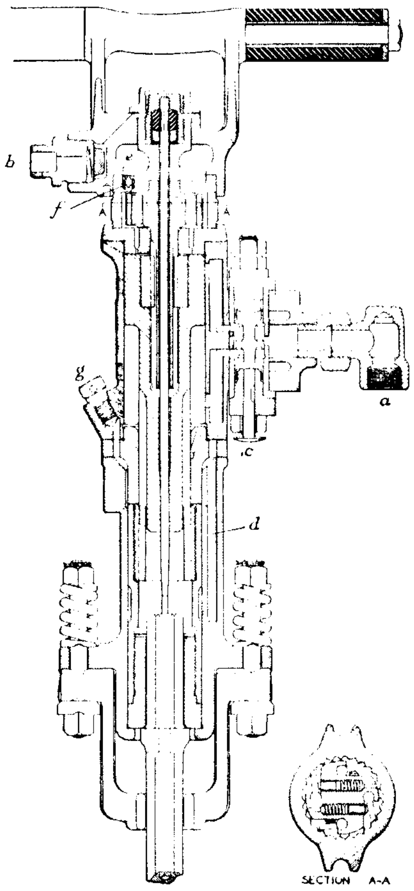


FIG. 45.—Sullivan L 7 Water-tube "Rotator."

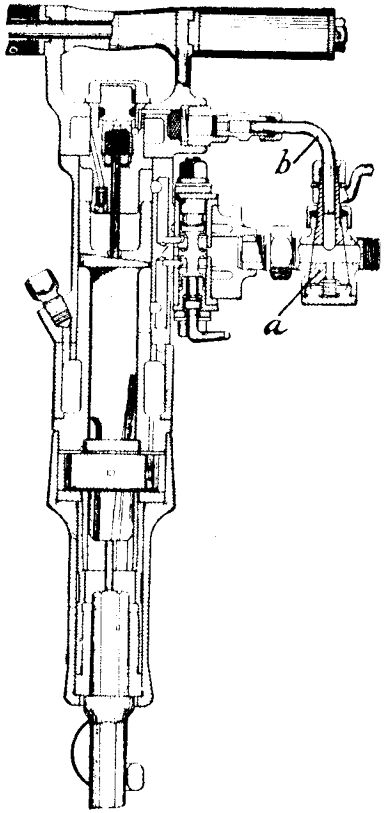


FIG. 46. Sullivan Steam-driven "Rotator," DP 331.

Chicago Pneumatic Tool Co. makes two sizes of sinker drill, CP-8 and CP 10, each in both dry and wet pattern, all having automatic rotation of the bit; also, an auger drill, CP-8A, with down-stroke rotation The CP-10W machine, as provided with water connection for wet drilling, is shown by Fig. 47. This company also makes two small, valveless drills for block-holing and other shallow work: K-13, weighing 29 lb. and drilling 3 to 6-ft. holes; No. 5, weighing 22 lb. and drilling 3 to 6-in. holes.

The sinker and auger drills have three-ringed, hollow spool valves, all of the same design, operating as shown in Fig. 48 (such of the main and auxiliary air ports as are not in the plane of this drawing are inclosed by areas of close hatching). The valve is indicated in its rear position, with the piston in its corresponding forward position and ready to begin the back stroke (to the right).

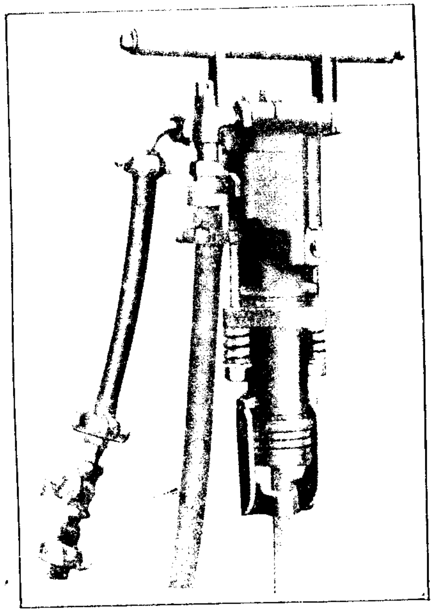


FIG. 47. Chicago Pneumatic Tool Co.'s CP-10W Sinker, for Wet Drilling.

Live air enters at *B* from the throttle, passes around the front neck of the valve to main port *C*, and, acting on the shoulder *E* of the piston, causes the back stroke. When auxiliary port *A* is uncovered by the piston, air enters through *A* and builds up pressure on valve face *L*. At the same time, auxiliary port *D* is covered by the piston, and, as the air previously acting against the valve's front face *F* is free to pass to the atmosphere through port *G*, the valve moves to the left. Live air entering at *B* then passes around the rear neck of the valve and through main port *H* to the cylinder, thus driving the piston forward.

When auxiliary port *D* is uncovered by the piston, air pressure acting on valve face *F* throws the valve again to the right. While the piston is moving forward, the air in annular chamber *E* passes to the atmosphere through ports *C* and *K*. Finally, as the air behind the piston is exhausted through port *H* and around the valve to port *J*, the valve then being in its rear position, the stroke is reversed, completing the cycle.

Gilman Manufacturing Co.'s hammer drills, brought out in 1922, differ from others in that the bit is rotated independently of the hammer strokes. In 1898, A. P. Schmucker, of Denver, Colo., designed a light drill on this principle. It was tested in the Cripple Creek and Leadville mines, but was not successful. Other attempts along similar lines followed. The chief difficulty seems to have been that the rotation motor was unbalanced, causing hurtful stresses and leakage of air.

The rotation of the Gilman drill appears to be entirely satisfactory. Instead of using the customary rifle-bar and splines, the bit is rotated by a balanced air motor, independently of the movements of the hammer. The speed of rotation is varied inversely to the bit's diameter, which insures proper reaming of the hole in rocks of different kinds. The peripheral speed of the bit can be varied between about 100 ft. per min. for soft rock to 50 ft. for hard rock. Incidentally, as the maker points out, the elimination of the cylinder spaces required by a

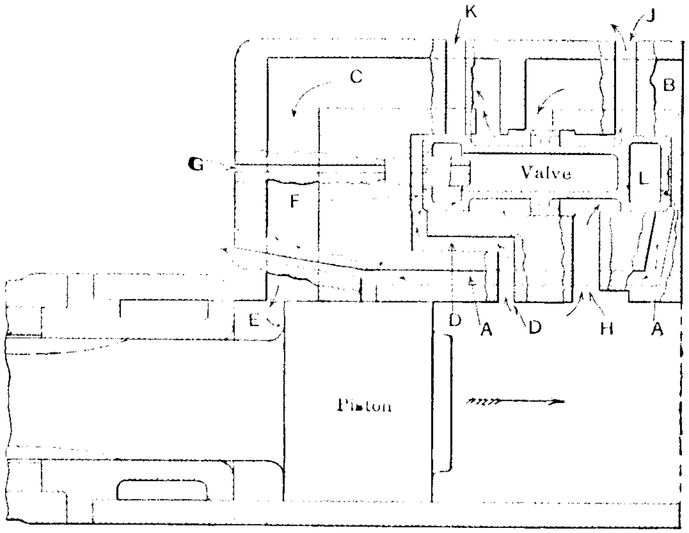


FIG. 48.—Valve Motion of Chicago Pneumatic Drill Co.'s CP-10 Sinker Drills (Diagrammatic).

rifle-bar and accompanying parts, which must be filled once per stroke with compressed air, reduces the consumption of air.

Fig. 48*a* is a longitudinal section of this drill, which is of the valveless type. Fig. 48*b* shows cross-sections through the rotary motor and gearing. The motor *a* (which may be either in front or rear of the hammer) is concentric with the drill cylinder, and is completely balanced in its operation. On its periphery are four large teeth *b*, 90° apart, which act as air pistons. The smaller teeth between the pistons engage with companion gears *c*, on two parallel shafts *d*. Revolving in the same direction, shafts *d* transmit power through excentrics (not shown) to the gyrating yokes *e*. These yokes have internal teeth, engaging at diametrically opposite points with the teeth on the chuck sleeve *f*.

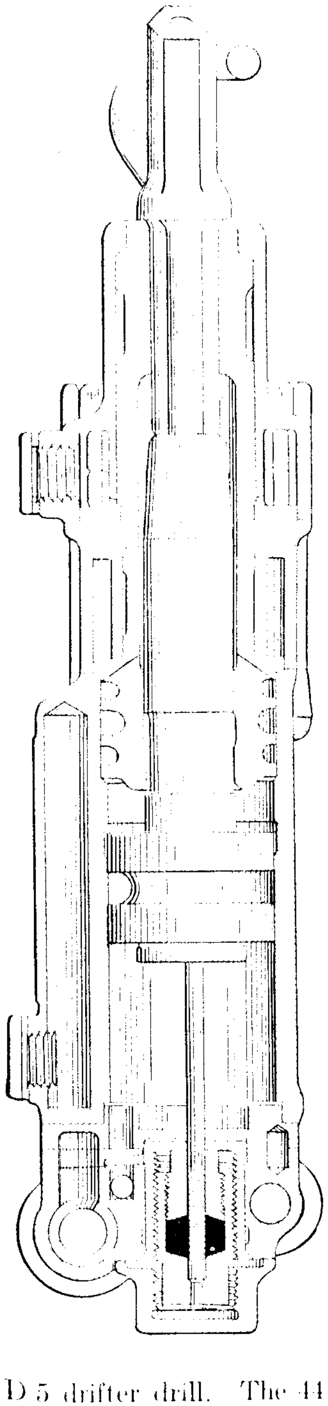


FIG. 48*a*.—Gilman Hammer Drill.

By this mechanism the rotation speed of shafts *d* is reduced to that required for the drill chuck. The speed relation depends upon the ratio of the numbers of teeth on the yokes and chuck sleeve.

These drills, with independent rotation, are made in 12 sizes, from 1¾- to 3¾-in. diameter of cylinder, and weighing 21 to 135 lb. In addition, there is

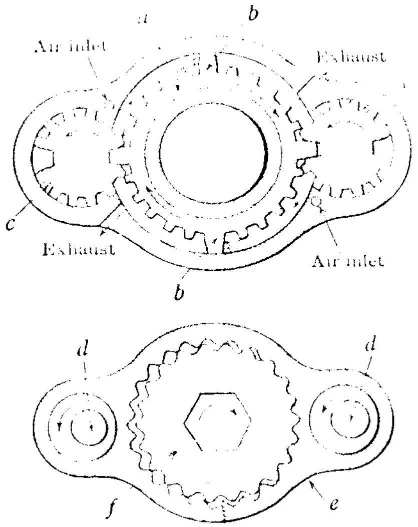


FIG. 48*b*.—Gilman Drill. Diagrammatic Sections of Rotation Motor and Gyrating Yoke.

a 2¾-in, 45-lb. drill, with rifle-bar rotation.

Cleveland Rock Drill Co. makes several types of hand hammer drills, as specified in Table IX.

Guide shells are supplied for mounting all the drills listed, when they are to be used for tunneling, drifting or breast stoping.

The **H 7** and **A 1** machines have three-ring spool valves, in a chest cast integral with the cylinder and operating like that of the Cleveland D 5 drifter drill. The 44 SD drills have a tubular or ring valve, set axially in the rear end of the cylinder. Its design is essentially the same as the valve of the D 8 drifter, details of which will be found under "Class A" hammer drills. In a special model of the 44 SD drill, a device is provided for extra strong blowing, for drilling in rock where it is difficult to keep the holes well cleaned. On stopping the exhaust, a cam opens a small relief valve, which causes the main valve and the piston to take their forward positions, thus allowing inlet air to pass directly through the hollow steel to the bottom of the hole.

TABLE IX. CLEVELAND HAND HAMMER DRILLS

	H 7 dry	H 7 W wet	A 1 auger	A 1 B dry and wet	44 SD dry	44 SDW wet
Cylinder diameter, inches	2½		2½		2⅝	
Stroke, inches	2½		1⅞		2¼	
Overall length, inches	22	23	19¼	20	20½	
Weight, pounds	18		35	39	43	44
Maximum depth of hole, feet	24		..	6–8	20	

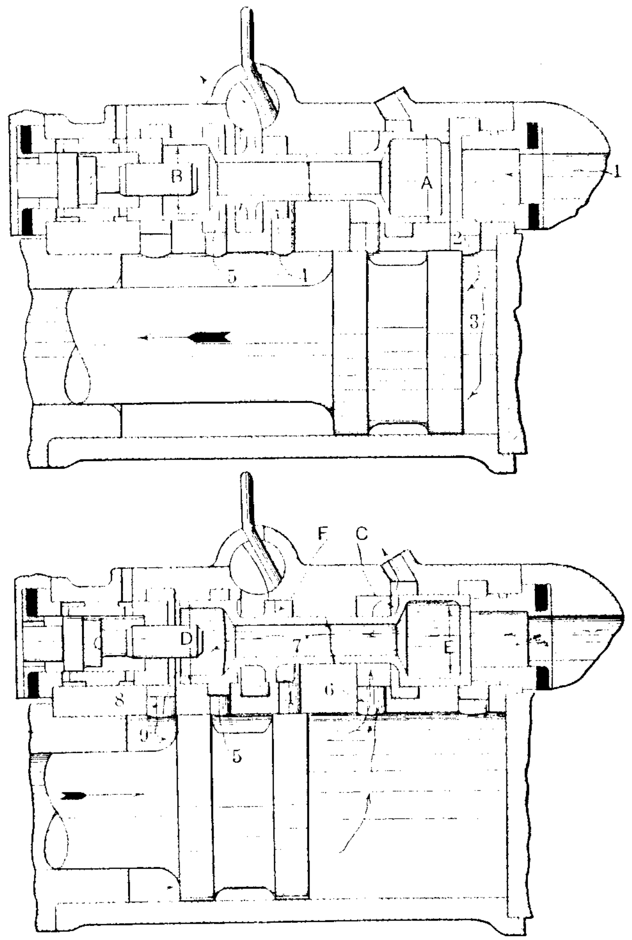
Gardner-Denver Co. makes the line of hand hammer drills specified in Table X. The designs are based upon those covered by the Waugh patents, formerly of the Denver Rock Drill Manufacturing Co., which is now merged in this company. Models 11, 37 and 93 are furnished in both dry and wet types. Model 16 VW, a light, valveless, hand-rotated drill using solid steel, is also made.

TABLE X. GARDNER-DENVER HAND HAMMER DRILLS

Model	11	37	90 (auger)	93	95 (dry)
Cylinder diameter, inches	2½	2⅝	2½		
Length overall, inches	21	22½	21¼		
Weight, pounds	50	65	41	42½	42½
Steel recommended	Hollow		Solid spiral	Hollow	
Size of steel, inches	⅞-1	⅞-1¼	1½	⅞-1	⅝-1

Air hose: for No. 11, ¾ in.; No. 37, ¾-1 in.; Nos. 90, 93, and 95, ¾ in. Water hose: for No. 11, ½ in.; Nos. 37 and 93, ⅜-½ in. Shells are supplied for mounting all these drills for use as drifters.

Model 37, also known as the "Turbro Waughammer," is especially adapted to heavy work, as in shaft sinking. The unique rotation device is independent of the hammer action. It consists of a small air turbine, which, through a worm reduction gear, causes the chuck to revolve. It has roller bearings, and the entire mechanism is inclosed in a housing between the cylinder and chuck. To avoid possible breakage, the chuck is driven by the turbine through a friction clutch. All the other drills listed in the table are provided with the usual ratchet and rifle-bar rotation.



FIGS. 49 and 50. Valve Action of Gardner-Denver Drill, Model 11.

Valves. Model 11 has a differential piston-valve of tubular form, placed in a chest on one side of the cylinder. It is made in two pieces, for convenience of assembly, but operates as a single unit. Figs. 49 and 50 show the action of this valve.

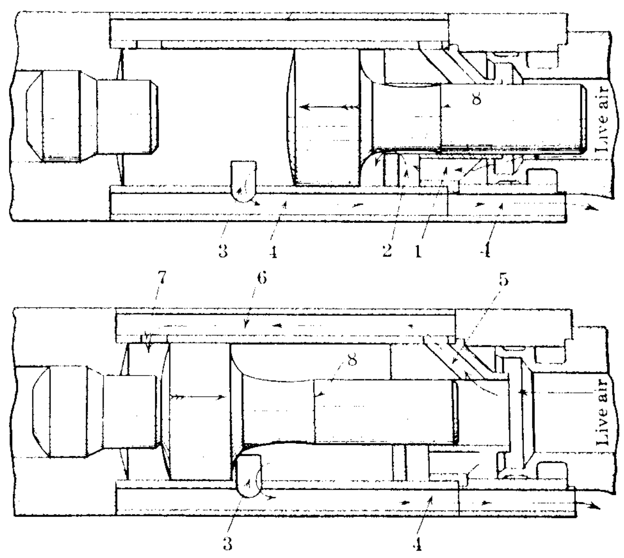
In Fig. 49 the hammer is at the beginning of the forward stroke. Live air entering from the throttle through 1, the valve is held in its forward position because its area at *A* is greater than its area at *B*. The air passes through port 2 into the rear end of the cylinder, 3, driving the hammer forward. Exhaust from the front end of the cylinder is meanwhile passing out through trip port 4. When 4 is covered by the advancing hammer exhaust still continues through auxiliary port 5.

Referring to Fig. 50, as the hammer uncovers rear trip port 6, live air acts on the valve face *C*. Since the combined area of *C* and *B* (see Fig. 49) is greater than area *A*, the valve is thrown to its rear position, and held there because its area *D* is greater than area *E*. Live air then passes through the valve's interior, 7, and by port 8 into the front end of the cylinder, where it acts on the annular face 9 of the hammer, driving it rearward. The rear end of the cylinder is then exhausting through trip port 6. When 6 is covered, and during the rest of the stroke, cushion pressure is being built up back of the hammer. Port 5 is now closed by the valve. When port 4 is uncovered, live air is admitted to the annular valve face *F*. As the combined area of *F* and *E* exceeds area *D*, the valve moves forward to the position in Fig. 49, completing the cycle.

Drill No. 11 has a special blowing device for cleaning the drill hole, in the form of a Corliss valve in the forward exhaust port. When this valve is closed by hand, the piston is forced to the front end of the cylinder and live air passes directly from the rear end into the hollow steel.

Model 37 is fitted with a tubular spool-valve of the differential type. Models 90, 93, and 95 have ordinary hollow, three-ring spool-valves. Model 16 VW is a recent, hand-rotated, valveless drill, the piston itself acting as a valve. Externally, it is similar to the Hardsocg No. 1 (Fig. 35), but the shape of the piston and its relation to the air ports are quite different.

Figs. 51 and 52 show the operation of the 16 VW drill. Live air acts constantly on the small rear end of the piston, and, during the forward stroke (Fig. 51) by way of ports 1 and 2, also on the back of the piston head. After shoulder 8 of the piston has shut off inlet air from the piston head, the stroke proceeds on expansion, plus the pressure of live air on the small end of the piston, until port 3 is uncovered, which exhausts the expanded air through port 4. In beginning the back stroke (Fig. 52), port 5 being uncovered by the rear part of the piston, live air enters by ports 6 and 7 to the front end of the cylinder. Port 5 remains open until the piston has delivered its blow and has made about one-quarter of its return stroke. This stroke is continued on expansion until port 3 is uncovered, allowing exhaust through port 4. Finally, when shoulder 8 uncovers port 2, live air is again admitted against the piston head. Thus, the drill is so designed that much of the work is done on expansion, the air being exhausted at about half the initial pressure.



Figs. 51 and 52.—Operation of Gardner-Denver Drill, Model 16 VW (Diagrammatic).

Holman Bros., Camborne, England, make a number of hand hammer drills, the principal types being specified in Table XI. Types LB and

TABLE XI.—Holman Hand Hammer Drills

	LB and WLB	PV 57	HD 1
Cylinder diameter, inches	$2\frac{3}{8}$	$2\frac{3}{4}$	$2\frac{3}{8}$
Stroke, inches	$2\frac{3}{4}$	$2\frac{1}{4}$	$2\frac{3}{8}$
Length overall, inches	24	$25\frac{1}{2}$	$18\frac{1}{4}$
Net weight, pounds:			
Dry	55	$57\frac{1}{2}$	$22\frac{1}{2}$
Wet	58		
Steel (hollow), inch	1		$\frac{7}{8}$
Air hose, inch	$\frac{3}{4}$		$\frac{3}{4}$
Water hose, inch	$\frac{1}{2}$		
Depth of hole, feet	15		12

WLB (dry and wet) and the PV 57 can be supplied with cradles, for mounting them on column or tripod. The small HD 1 drill is made in dry form only. Two other light-weight drills are the PV 21, used chiefly in coal and soft rock, and the PV 31, applicable to general service; their weights are respectively 21 and 41 lb. All these machines have automatic rotation of the usual design.

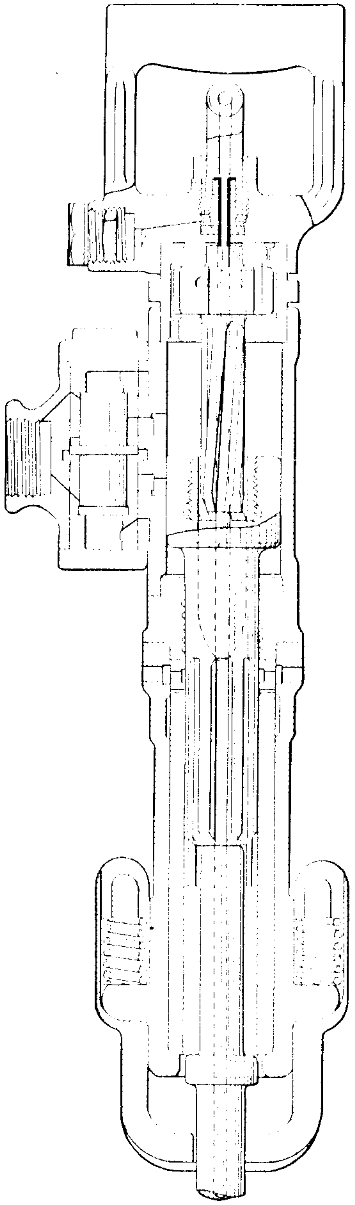


FIG. 53.—Holman Wet Drill, WLB.

The PV 57 drill is fitted with the Holman "flap" valve, exactly the same in operation as that used in the drifter drill PV 160, for details of which see Fig. 20 and accompanying description. Drills 21 and 31 of the PV type are also fitted with the flap valve, but in a modified form. In the former, the relief valves under the main valve are omitted altogether, thus making necessary only two auxiliary ports instead of four.

Fig. 53 shows the general features of the LB drill. The valve is a solid differential spool, having a simple action that is quite different from that of the spool valves already described. Its operation is shown by Fig. 54. The piston *A* being in its striking position, the valve *B* admits air to the forward end *D* of the cylinder, by way of ports *E* and *F*. When the piston on its back stroke uncovers port *C*, air from the cylinder acts on the left-hand annular face of the central valve ring. As this has a larger area than the end of the valve, the latter is thrown backward, thereby closing ports *E* and *F*, and admitting inlet air to *H* and through port *I* to the rear of the cylinder, *K*. The air in the forward end of the cylinder is then exhausted to the atmosphere through port *Z*, and the stroke is reversed by the live air entering through port *I*. Air now acts on the right-hand side of the central valve ring, throwing the valve forward to its position as shown. Since the piston covers port *I* when port *J* is open, and vice versa, positive action of the valve is insured, as live air can not enter both ports at the same time.

Climax Rock Drill and Engineering Works is located at Carn Brea, England. Specifications of the drills of this company that fall in Class B are given in Table XII. These machines are designed for both dry and wet drilling, and have automatic rotation of the conventional type. Except the Joburg No. 3, all may be supplied with cradles for mounting on columns. Besides those in the table there are three smaller sizes, for dry drilling only: the Britannia and the F1 and F2, the latter weighing respectively 19 and 25 lb.

TABLE XII. CLIMAX HAND HAMMER DRILLS

Model	Cylinder diam., in.	Stroke, in.	Weight, lb.		Length overall, in.	Blows per min.	Cu. ft. free air at 80 lb.
			Bare	Mounted			
Joburg No. 1	2⅜	2	46	102	20¼	1,000	60
Joburg No. 2	2⅜	2½	60	120	22¾	1,000	65
Joburg No. 3 sinker	2¾	2 9/16	120	......	29¼	950	110
J 114	2⅜	2½	60	130	22¾	1,550	95
CX sinker	2¾	2½	60	115	22¼	1,760	95

Spool valves are used in all Climax drills. The Joburg models have solid, two-ring spools; J 114 a single-ring spool; and the CX a two-ring hollow spool. The valves of the Britannia and F models are hollow, and set transversely on the cylinder. Following are examples of the valve actions.

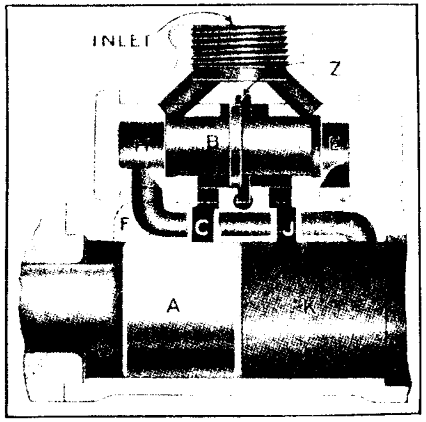


FIG. 54.—Valve Action of Holman Drill WLB.

Fig. 55 is a diagram of part of the Joburg No. 3 drill. Live air, entering from the throttle through the main port, acts constantly in the recesses *F1* and *F2* encircling the valve grooves. From these air passes constantly through small holes (shown by dotted lines) in the valve rings to both ends of the chest. Auxiliary ports *R1* and *R2* lead from chest to cylinder. One or other of these ports is always covered by the piston, and, when either is thus closed, the valve moves to the opposite end of the chest. With the piston in the position shown, live air passes through *F1* to *P1* and thence by main port *RP* to the rear of the piston, causing the forward stroke. When auxiliary port *R1* is uncovered and *R2* closed, the valve moves to the right, shutting off air from the rear end of the cylinder and admitting it to the front end; the latter having previously been exhausting through main port *FP* to *P2* and port *E*, which is in communication with the atmosphere. In making its throw, the valve must move about $\frac{3}{16}$ in. before any live air can enter the cylinder; that is, before *F2* is brought into communication with *P2* and *FP*. During this small movement of the valve, the piston will continue its left-hand stroke and so uncover exhaust port *RE*. On the return stroke, when *RE* is again closed, the remaining air in the cylinder's rear end exhausts through port *E*. Some exhaust from the front end

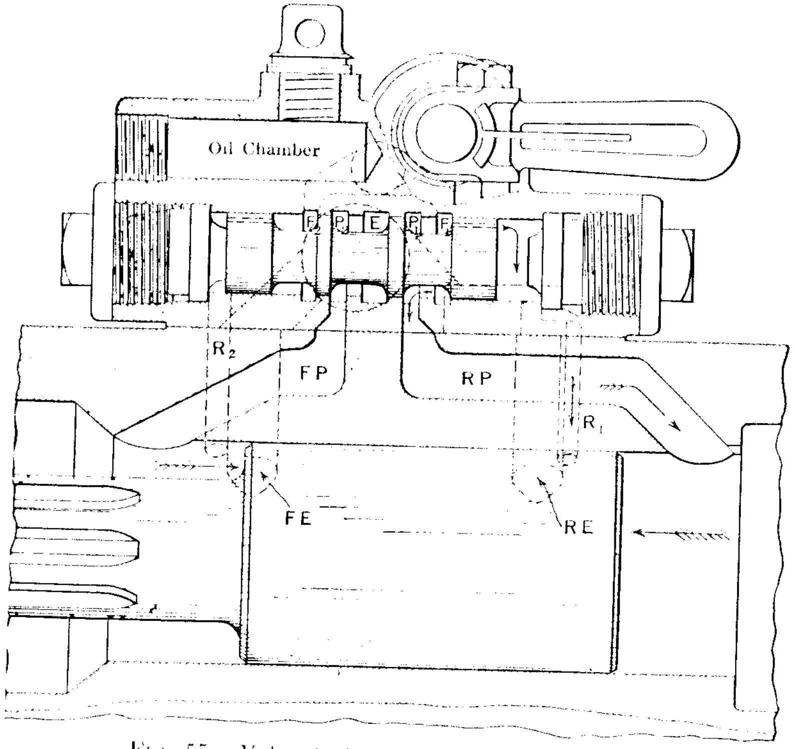


Fig. 55.—Valve Action of "Joburg No. 3" Drill.

escapes along the grooves on the piston shank to the drill chuck, mixing with the water jet passing into the hollow steel.

Fig. 56 shows the valve action of the CX drill. *A* is the inlet, *B* the exhaust. Live air enters constantly to grooves *C* and *D*, near the ends of the valve. The piston is about to make its back stroke. When groove *H* is uncovered in this stroke, live air passes by way of auxiliary port *I* (see dotted lines) to the right-hand face of valve-ring *J*, thus throwing the valve to the left. At its midstroke, main port *P* is closed and *K* opened, thus admitting air to the rear end of the cylinder and causing the piston's forward stroke. As the piston strikes the bit, live

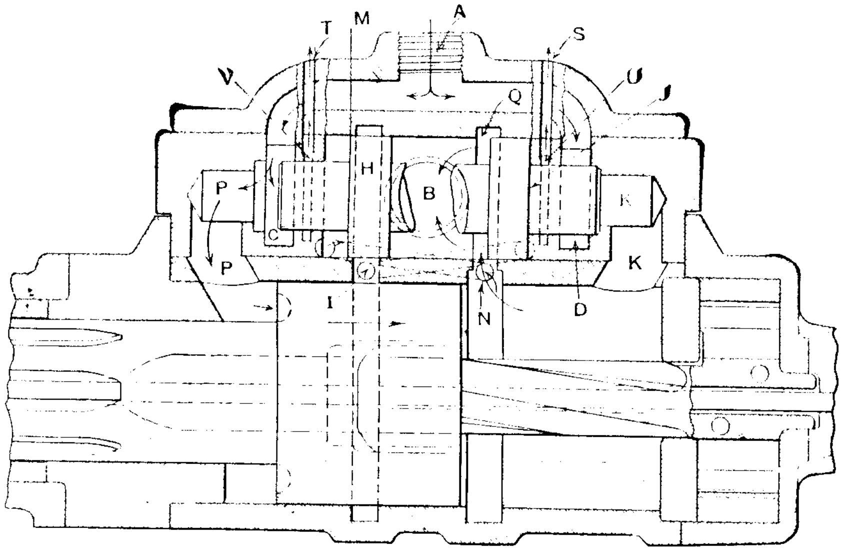


Fig. 56.—Valve Action of the Climax CX Drill.

air from the cylinder passes through *N* and by the dotted port to the left-hand face of valve-ring *M*. When the valve is again at midstroke, air passes through groove *C* and port *P* to the cylinder, also uncovering groove *Q*; thus completing the cycle. Vent ports *S* and *T*, from grooves *U* and *V*, entrap and exhaust to the atmosphere any air leakage from *C* and *D*, which otherwise might interfere with the valve's proper operation.

A simple, oscillating valve, of tubular form, is used in the Climax Britannia and F drills (Fig. 57). It is contained in a cage-like chest, is very light and has a small amount of movement between the valve seats *C* and *C1*. When the piston is at the end of its back stroke, seat *C1* is uncovered, thus allowing air to flow from space *F1* through port *G* into the rear end of the cylinder, and causing the forward stroke. After the piston has passed over and uncovered exhaust port *E*, the pressure in space *F1* falls, so that the live air in *F* throws the valve to the right. This closes seat *C1* and uncovers seat *C*, admitting air through port *I* to the front end of the cylinder. Then, as the piston moves rearward, it uncovers exhaust port *D*, out of which passes the live air in space *F*, thus reversing the valve to its first position, as in the cut.

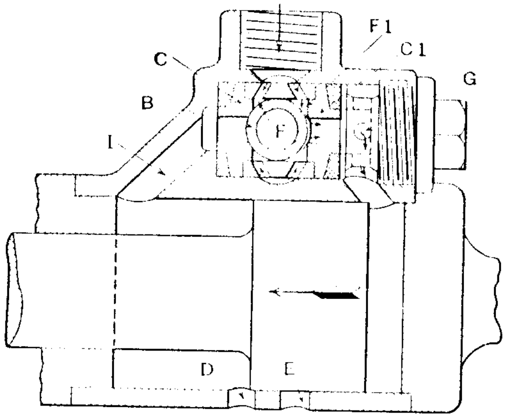


FIG. 57.—Oscillating Valve, F Type of Climax Drills.

Flottmann Aktiengesellschaft,* Herne, Germany, makes several types of hand hammer drills (Table XIII), embodying differences in design from those already described.

TABLE XIII. SPECIFICATIONS OF FLOTTMANN DRILLS

	AZ 11	AZ 16	AZ 25	N 55	N 55 (light)
Cylinder diameter, inches	$2\frac{1}{8}$	$2\frac{3}{8}$	$2\frac{9}{16}$	$2\frac{1}{6}$	$2\frac{1}{6}$
Weight, with steel holder, pounds	26*	35	55	35‡	31‡
Steel shank, inches	$\frac{3}{4} \times 3\frac{3}{16}$		$\frac{7}{8} \times 4\frac{1}{4}$†	18.5 × 80 mm. = abt. $\frac{47}{64} \times 3\frac{5}{32}$	

* Weight, without steel holder, 21 lb.
† For hollow steel, shank is 1 in. hexagon × $1\frac{1}{4}$ in.
‡ Equipped for wet drilling, the N55 drills weigh respectively 38 and 34 lb.

The AZ models (Fig. 58) are made with solid or hollow pistons, for dry drilling only; the N drills are designed with solid or hollow pistons for both dry and wet drilling. The company also makes model AUL, a heavy sinker drill without automatic rotation, weighing 66 lb., and model AN 65, resembling in general features the drills listed in the table, weighing: solid piston, 53 lb.; dry blowing, 58 lb.; wet, 60 lb. For mounting on a light column, for drifting and similar work, the Flottmann drills, except the hand-rotated model AUL, are furnished with a saddle or shell, as is usual with nearly all Class B drills of other makes.

The rotation mechanism differs from the usual design, shown by Fig. 15. The straight longitudinal flutes, engaging with the chuck, are on the forward end of the rifle-bar; and the ratchet, instead of being in the rear end of the cylinder, encircles the shank of the rifle-bar, in front of the piston head. There are therefore fewer moving parts.

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* This Company is affiliated with the British Flottmann Drill Co., Cardiff, Wales.

Valves. Model AUL has a light-weight spool valve. Models N and AN 65 have single ball valves, the operation of which resembles that of the tubular valve of the Climax F drill (see Fig. 57) and of the ball-valve formerly used in the old Hummer drill of the Chicago Pneumatic Tool Co. (See the previous edition of this book.)

Model AZ is unique in being equipped with a pair of ball valves, set side by side in the cylinder casting, each in its own chest, and moving synchronously (Fig. 59). Above the valve chests is a common air inlet. Operating exactly like the single ball valve, the twin balls have the advantage of being lighter and, with a smaller lift, each controls the same total port area. Having less inertia, their wear is reduced and the speed of the drill can be increased. Air vents near the cylinder ends prevent excessive compression on both strokes of the piston.

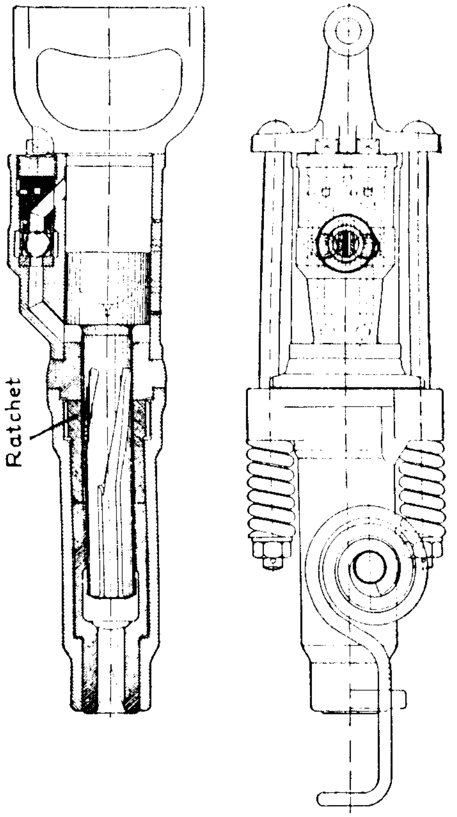

FIG. 58.—Flottmann Twin-valve Drill, Model AZ.

Hand Pick Machines.—Designed on the general lines of Class B hammer drills is a group of tools known as picks, concrete and pavement breakers, spaders, etc., used for surface excavation. They differ from true drills chiefly in having pointed or flat bits and requiring no rotation mechanism.

The principle of the hammer pick, as applied to mining is not new. A pick of this kind was used in Europe, for stoping in thin veins, nearly forty years ago. In recent years the idea has been revived by several drill manufacturers. Hammer picks are now employed advantageously in mines as auxiliary tools, for dressing the walls of workings, taking down partly loosened ore or rock after blasting, breaking large pieces, cutting drains in gangways and drifts and making hitches for timbers. In coal mining they have tended also to replace the heavy mounted picks for shallow undercutting and for increasing the height of deep undercuts made by chain and other machines (see under Coal Picks, Chapter XXII).

Experience with hammer picks in the Nourse Gold Mines, South Africa, where a number of them (made by the Consolidated Pneumatic Tool Co., of Great Britain, affiliated with the Chicago Pneumatic Tool Co.) have been used since 1927, in connection with machine-drill work, shows favorable results in reducing the cost of labor.

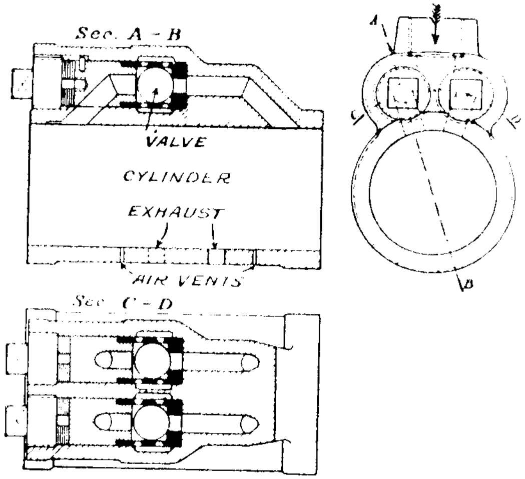

Fig. 59.—Flottmann Twin Ball-valves (Diagrammatic).

Class C. Stoping Drills

These machines are intended primarily for drilling holes directed steeply upward, as, for example, in overhand stoping. They are often called "stopers" or "stopehammers." A fundamental feature of their design is that they have, attached to the rear end of the drill cylinder, a long telescopic air-feed standard, by which the drill is supported while at work, and which automatically keeps the bit fed forward against the bottom of the hole. Anvil blocks are almost invariably used.

Valve Action.—The valves are generally of the same types as those used by the different makers for their drills of classes A and B. A few stoper drills are valveless, the functions of the valve being performed by the movements of the piston (see the valveless drills cited in Class B). To avoid repetition in the following examples, the kind of valve will be merely named, or very briefly described, operating details having been already given.

Rotation.—Most drills of Class C are rotated by the operator, a right-angled handle being screwed into the cylinder casting. Certain of the later types (since about 1913) of the Ingersoll-Rand, Sullivan and Gardner-Denver drills have automatic rotation, the details of which

are the same as for Classes A and B hammer drills, as already described.

A few of the stoper drills are provided with means for mounting them on the arm of a light screw-column, or on a tripod. They can then drill holes approaching the horizontal, as in drifting, breast-stoping and similar work. The attachment to the column consists of a ring clamp encircling the cylinder of the feed standard.

Air Feed.—This consists of a long, slender cylinder and piston, attached to the rear of the drill-head. Into the lower end of the feed piston is screwed a heavy stud, with a conical point, upon which the machine stands on the floor of the stope or other working. When the drill is in operation, compressed air is admitted to the upper end of the feed cylinder. The air pressure, reacting against the piston, forces the cylinder upward, carrying with it the drill-head, and thus automatically keeping the bit in contact with the bottom of the hole. The maximum length of "run" of the feed piston generally ranges, in the different makes of drill, from about 18 to 25 in. In most stopers, the admission of air to the feed cylinder is under control of the operator, thus enabling him to adjust the feed pressure to the drilling conditions of the rock.

For air-feed stoping drills, automatic rotation is not so clearly advantageous, nor so generally applied, as is the case with the drifter and sinker types of drill. The chief distinction is that the air-feed of the stoper exerts a constant pressure on the bit against the bottom of the hole. Therefore, in non-homogeneous or fissured rock or ore, automatic rotation often fails to work satisfactorily; the bit may stick fast ("fitcher") under a uniform feed pressure. This difficulty is overcome by providing a "push-button" or other simple device, convenient to the hand of the operator, by which the air pressure in the feed cylinder can be instantly reduced, whenever drilling conditions make such control desirable or necessary. This point should be clearly understood, as it is sometimes assumed that the usual non-controlled automatic rotation is always successful, and that any stoper drill having it can be mounted on a column for drifting or other horizontal work.

Aside from the automatic air-feed, stoper drills generally differ from Class B drills of the same make in no essential particular. Therefore, only a few examples are presented briefly.

Ingersoll-Rand "Stopehamers."—To the beginning of the year 1929, the hand-rotated machines were designated as BC, CA, CC 1 and CC 2; the automatically-rotated as R 51. The "C" drills include sundry variations of ordinary dry and wet machines, the drill-heads being substantially the same as in the "Jackhamers": wet drills with hand-controlled air-feed and two concentric axial tubes, for admitting both air and water to the bit; and certain special wet designs having either an ordinary single tube or no water tube—in the latter the water being admitted through the front head of the cylinder.

The company has now (1929) concentrated upon five types of "stopehamer": the CA, CC and N-38, rotated by hand, and the R-51, which, with two new "Jackstopers," R-39 and S-49, have automatic rotation. The R-39 and S-49 machines are modifications of the "Jackhamers" having the same type designations (see Fig. 38 under Class B drills). Specifications of all the current "Stopehamers" are given in Table XIV.

TABLE XIV.—INGERSOLL-RAND "STOPEHAMERS"

	CA 31	CC 1	CC 2	N 38	R 51	R 39	S 49
Cylinder bore, inches	1¼ by 2¼	1⅜ by 2¾	1⅜ by 2¾	2 11/16	3	2⅝	2⅝
Stroke, inches	3⅛	2⅞	4	3¼	2 11/16	2	2½
Length, feed closed, inches	51½	51	54	54⅜	61½	54½	55¼
Length, feed extended, inches	75½	72¼	75½	76⅛	86	79	81¼
Total feed travel, inches	24	21¼	21¼	21½	24½	24½	25½
Diameter of air-feed cylinder, inches	2	2¼	2⅜	2¼	2 9/16	2	2
Weight, pounds	62½	81½	93½	97½	120	88½	95

NOTE.—The air inlet of all these machines is ¾ in.

Fig. 60 shows the CC-2 dry drill, which has been used successfully for many years; *a* is the hammer, *b* the anvil block and *c* the air inlet. The simple cylindrical valve *d* has a very short throw, end seats, large ports, and the drill is designed to work with a high degree of expansion. To minimize dust nuisance, due to agitation of the dry cuttings, the exhaust is directed backward. The air-feed is described later. In the BC drill, the "Butterfly" valve is used (see Figs. 40 and 41 and accompanying text). Both of these drills are made in several forms and sizes.

The N-38 drill (Fig. 61) has an ordinary single water tube, like the "Jackhamers." It can be used, however, as either a dry or wet machine. The valve is of the "flapper" type (see Fig. 18) and the hammer is a plain cylinder. Front heads are furnished for hexagon, quarter-octagon, square, or cruciform steel. As in the CC drills, the exhaust is directed backward. In the front head is a vent hole, to permit escape of dirt that may enter the chuck around the bit shank. Lubrication is entirely automatic.

In the R-51 "stopehamer" (Fig. 62), blowing is done by the double

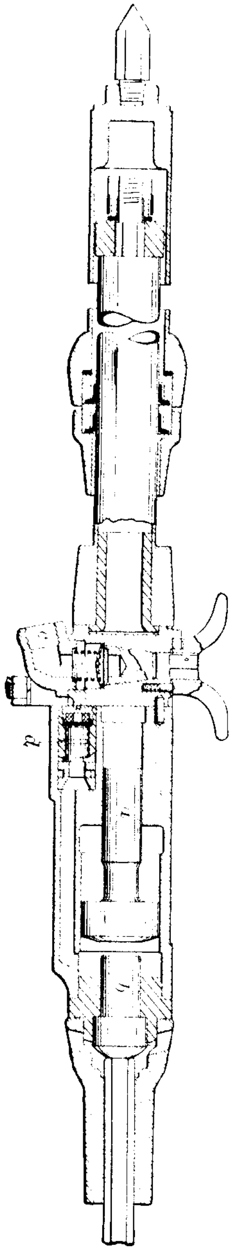

FIG. 60.—Ingersoll-Rand "CC Stopehamer"—Dry Type.

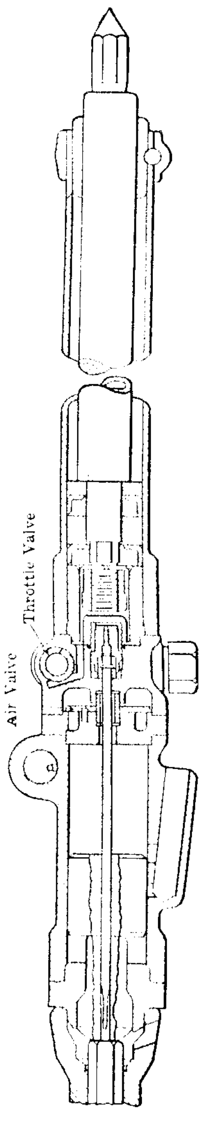

FIG. 61.—Ingersoll-Rand N-38 "Stopehamer."

axial tubes, referred to above. Water passes through the inner tube and is joined, on entering the hollow steel, by compressed air admitted to the annular space between the two tubes. The valve is of the "flapper" type (for details, see Fig. 18, under drifter drills, and accompanying description). There is also a small spool valve in the side of the cylinder, which relieves the air cushion at both ends of the piston travel, and gives a quick exhaust. The air inlet throttle also controls the admission of water, so that both are turned on and off together. This prevents possible flooding of the drill with water, which would wash out the lubricant.

The air-feed for the "stopehamers" is designed in three ways: (*a*) plain type, as noted in the prefatory remarks on stoper drills; (*b*) extension type, in which the length of the pointed stud attached to the feed piston-rod can be adjusted according to the varying distance between floor and roof of the working place; (*c*) reversed type, in which the feed piston-rod is attached to the drill-head. Design (*c*) is employed when the machine is to be mounted on a column or tripod, for drifting or similar work; in which case, the feed cylinder is clamped to the mounting. To prevent leakage of air, the piston of the feed cylinder has two cup-shaped leather washers (see Fig. 60), the air pressure forcing them against the cylinder walls.

Holman Bros., Camborne, England, make the wet stoper drill shown by Fig. 63. Specifications: cylinder diameter, 2 in.; stroke, 5 in.; weight, 86 lb.; length, 58 in., or with feed extended, 80 in.; air consumption at 80 lb., 60 cu. ft. free air per minute; blows per minute, 2,000.

The valve is a light, differential spool, placed axially at the rear end of the cylinder, as shown by the heavy black lines in the cut. Its movements are controlled by the reciprocations of the plain cylindrical hammer. This drill has no central tube; water is fed through a hose connected to a water-head surrounding the anvil block. The head has three annular grooves, two of them with rubber packing to prevent leakage of water into the cylinder. From the middle groove the water passes through a hole in the anvil block to the hollow steel.

The air-feed piston is attached directly to the drill-head, the cylindrical shell sliding upon it. In the end of the shell is set the pointed stud upon which the drill stands while at work. Compressed air is admitted to both drill-head and feed cylinder by a single hand-operated valve, on the side of the feed cylinder.

By a throttle lever, a "single-way" tumbler is turned for adjusting admission of air to the spool valve. The tumbler is of conical form and is kept tight in its seat by a spiral spring.

When the stoper is mounted on a column (Fig. 64), a clamp firmly

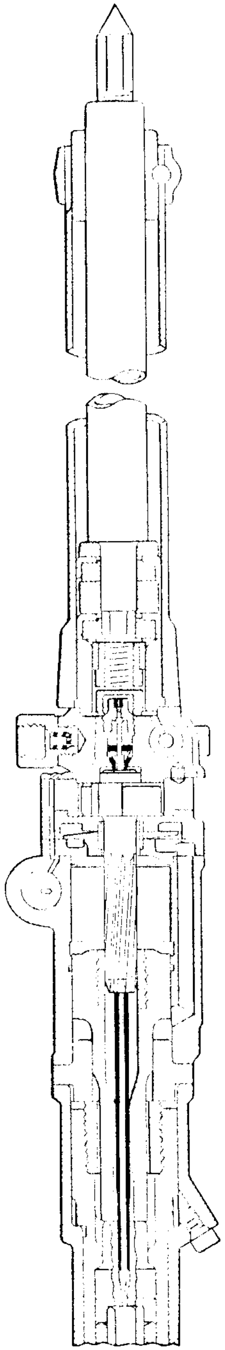

Fig. 62.—Ingersoll-Rand R-51 "Stopehamer."

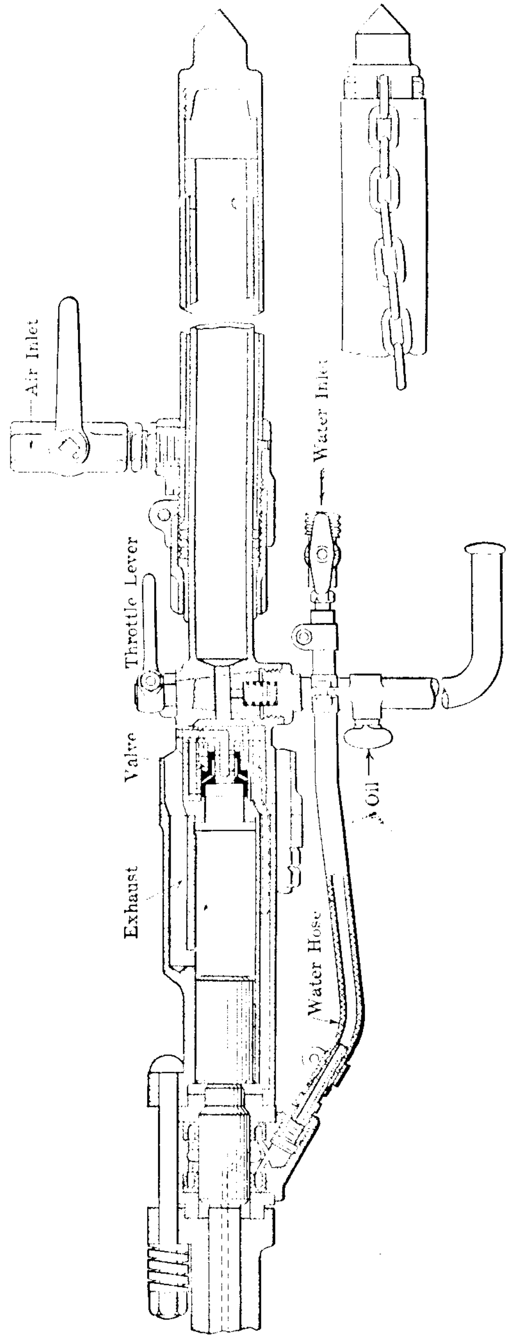

Fig. 63.—Holman Bros.' Air-feed Stoper AFW 2.

TABLE I. SULLIVAN AIR-DRIVEN CHAIN MACHINES

	CLA 2	CE 7
Length of machine body, inches	93¾	90
Width, inches	30	32
Height, inches	12*	24½
Height of undercut (kerf), inches	5 or 7	5¾
Length of chain-bar (jib), inches	30 to 90†	66 to 90
Net weight (less jib), pounds	1,075	5,800
Weight of 4½-ft. jib, pounds	572	
Diameter of feed-rope drum, inches	16	(chain)
Drum capacity, feet of ½-in. rope	90	....
Speed of cutting feed, feet per minute	15 to 30	15 to 30
Air pressure, pounds	30 to 70	20 70
Horsepower of air motor (average)	30	30

* When cutting 7 in. kerf, height of machine is 14 in.

† Lengths of the standard jibs available vary by intervals of 6 in. from 30 to 90 in., with an 8-ft. special jib.

the face of coal, the frame or jib carrying the cutter chain lies close to the floor, the main sprocket being then underneath the driving gearing. In another form of the machine (Fig. 8), for making an "overcut" 2½ to 3 ft. above the floor of the seam, the sprocket is placed above the driving gear. The same illustration also shows how the entire machine can be raised above its bedplate by four jack-screws, to make an over-cut in any part (within limits) of the face of coal.

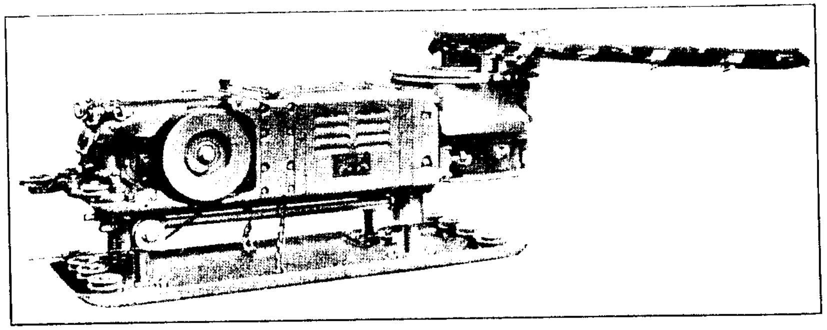

FIG. 8.—Mavor & Coulson's Air-driven Chain Machine (with set-up for "overcutting").

This coal cutter, widely used in Great Britain and elsewhere, is made in two sizes (Table II). It will be seen from its dimensions that it will work in very thin seams.

Rotary-bar Cutters were formerly used to a limited extent in the United States for room mining, but are no longer made here. In Europe, also, they have been largely superseded by chain machines.

TABLE II.—MAVOR & COULSON'S AIR-DRIVEN CHAIN CUTTER

	Model AXN	Model BXN
Length, inches	98½	108
Height, inches	15	19¼
Width, inches	30½	33¼
Net weight, pounds	1,800	6,770
Air pressure at machine, pounds	15	45
Cubic feet free air per minute, normal pressure, full load	400	600
Horsepower at normal pressure	25	35
Depth of undercut, inches	42 to 54	42 to 72

Maximum depth of cut depends upon the hardness of the coal or other material.

Fig. 9 shows the operating parts, with the casing removed, of the latest form of bar cutter for longwall mining, built by Mavor & Coulson. It is driven by an air turbine, similar in principle to that in Figs. 4 and 8. There are three sizes, the specifications of which are given in Table III. In this improved design, the turbine rotors are provided with axial and radial adjustments to maintain proper clearance and to take up wear.

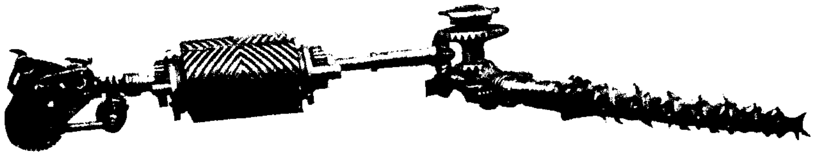

FIG. 9.—Operating Parts of Mavor & Coulson's Bar Cutter.

The tapering cutter bar has a series of staggered bits, set in sockets throughout its length, and is threaded spirally, to draw out the cuttings from the groove as the machine advances along the face of coal. The bits are readily removed for sharpening and replacement.

Like other coal cutters, this machine may be mounted on runners or skids for sliding on the floor of the seam, and can be moved on a track from place to place by means of a low, light truck. To groove a seam at a distance above the floor, the cutter is stood on a base-frame, which raises it sufficiently to make the cut at a height of say 4 or 5 ft.

In longwall work the cutter pulls itself along the face, as already described, by a wire rope or chain winding on a small drum attached to the machine, the drum being driven by gearing from the turbine motor. The cutter will thus operate in steeply-pitching seams, traveling automatically up or down the dip (though it is rarely desirable to cut downhill). For pitches exceeding say 28°, a safety rope is provided (Fig. 10). This rope *a*, anchored at each end of the face, makes two turns around each of a pair of pulleys *bb* on the outer side of the machine. By means of ratchets, these pulleys are free to turn only in the direction corresponding to the advance up the slope of the seam. If, on a steep pitch, the pulley rope *c* should break or its anchorage give way, the machine is held by the safety rope *a*. With this device, the cutter has operated on pitches as steep as 70° or even 80°.

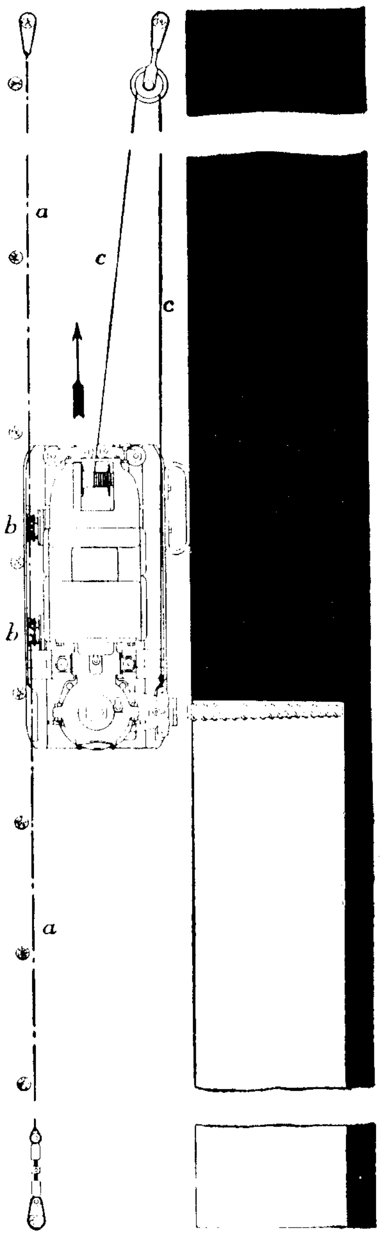

Fig. 10.—Mode of Operating a Bar Cutter on a Steep Pitch.

Disk cutters are a very old type, invented in Great Britain, and still occasionally used there and on the continent when the conditions are suitable. They are entirely obsolete in the United States.

Designed for longwall mining only, disk cutters are useful in thin, steeply pitching seams, being fed along the face by their own power, as already described under the other types of cutters. They are especially adapted to undercutting hard or boney coal, but their field of service is quite definitely limited by certain physical characteristics of the coal seam and its roof rock. They should not be used: (1) in friable coal; (2) where the roof bends or caves easily; (3) in short-faulted seams; (4) where the undercut is to be made more than a few inches above the floor. Due to their design and mode of operation, they are more liable to be subjected to severe shocks and stresses than other cutters. A slight settling or collapse of the face of coal may jam the cutter disk; or, on account of the large diameter of the disk, if a nodule of pyrites or other hard substance be struck, the relatively deep penetration (depth of bite) of the teeth may cause a heavy shock to the entire machine.

Table III.—Mavor & Coulson's Air-driven Bar Cutter

	Model KN	Model AN	Model BN
Length, inches	84	86	97
Height, inches	13½	15⅞	19½
Width, when cutting, inches	38	39½	44½
Width, when moving, inches	30	30½	30½
Net weight, pounds	2,900	4,140	5,630
Air pressure at machine, pounds	45	45	45
Cubic feet free air per minute, normal pressure, full load	300	400	600
Horsepower at normal pressure	15	25	35
Depth of undercut, inches	36 to 42	42 to 54	42 to 72

Maximum depth of cut depends upon the hardness of the coal.

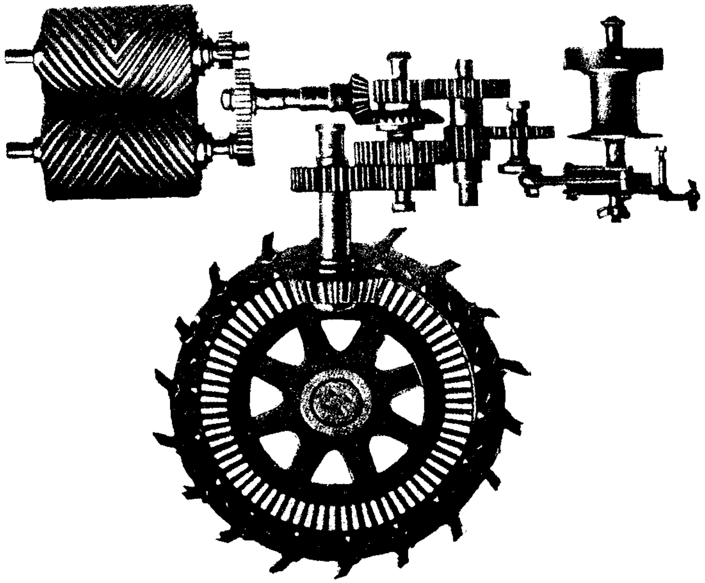

Fig. 11.—Operating Mechanism of the Mavor & Coulson's Disk Cutter.

The cutter disk is carried by a bracket on the frame, from which it projects a distance nearly equal to its diameter. On the periphery are set staggered bits, cutting a groove that admits the disk freely; speed of the disk, 15-30 rev. per min., depending on the character of the coal.

Fig. 11 shows the mechanical details of the air turbine-driven Mavor & Coulson disk cutter (first built with air drive about 1914). For specifications, see Table IV. It is also made with electric drive. The train of gearing, from the turbine to the cutter wheel carrying the teeth or bits, somewhat resembles the "transmission" of an automobile, and is shown so clearly that no description is necessary. All working parts, except the cutter wheel, are completely inclosed, as with other coal cutters.

TABLE IV.—SPECIFICATIONS, MAVOR & COULSON'S DISK CUTTER

Length, inches	99
Height, inches	16⅜
Width, when cutting, inches (space required in front of face)	42
Total width, including disk, inches	76
Net weight, pounds	5,700
Air pressure, full power, pounds	45
Horsepower at normal pressure	35
Free air; full load, normal pressure, cu.ft. per minute	600
Normal depth undercut, inches	42 to 48

This machine will cut normally at a linear speed of 18 to 24 in. per minute, and a set of bits should last at least for grooving 15 to 20 linear yards along the face, before resharpening is required. The working conditions to which the machine is best adapted have already been stated.

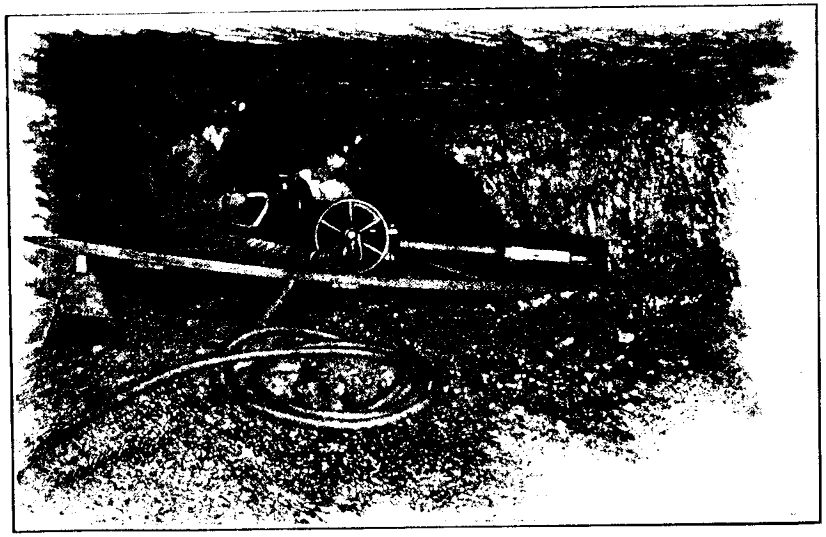

FIG. 12.—Mounted Coal Pick, Working in a Thin Vein.

Pick Machines are of two distinct forms: mounted and unmounted. They are much lighter, simpler and less expensive than coal cutters of the first three classes, and have a special advantage in being able to work around and loosen hard nodules of pyrites, etc., instead of cutting through them.

Fig. 12 shows a wheel-mounted pick, a type that has been widely used for many years, but which is now largely replaced by hand-held hammer picks. The cylinder and its appurtenances have a general resemblance to those of a piston rock-drill, except that there is no rotation of the bit. The machine stands on a wooden platform, sloping towards the face, so that the recoil of the blows is nearly neutralized and the machine is automatically kept up to its work. The operator directs the blows by swinging the machine laterally, with the wheels as a fulcrum. To give sufficient reach, the front cylinder head and piston rod are very long, commanding from one set-up a horizontal width of 4 or 5 ft. of undercut, by a depth of 3½ to 5 ft. The height of undercut tapers from 12-14 in. at the face to 3 or 4 in. at the bottom. A helper clears away the débris with a long-handle shovel, and assists in moving and setting up the machine.

The valve mechanism of a typical wheel-mounted coal-pick is shown by Figs. 13 and 14. Referring to Fig. 14, the main ports S, S^1 are controlled by slide-valve G, on the back of which is a lug H, engaging spool-valve F. Air enters alternately the opposite ends of F through ports J, J^1, which

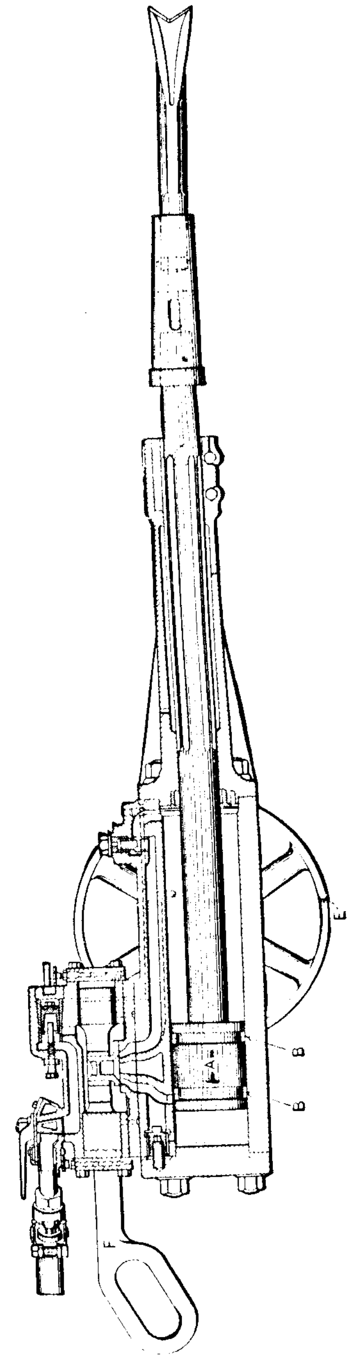

FIG. 13.—Ingersoll-Rand Coal Pick.

are controlled by auxiliary slide-valve *K* and its spool-valve F^1. Valve F^1 is operated by ports *N*, N^1, N^2 and N^3, which connect on either side with main ports *S*, S^1. Thus the rear end of chest of F^1 is connected with forward main port *S* and the forward end with rear main port S^1; when air enters the cylinder through S^1, a little of it passes through *N*, N^3 and throws valves *K* and F^1. Air entering through port J^1 reverses valves *F* and *G*, thus opening *S* to live air and S^1 to the exhaust. Some air passes from *S* through ports N_1, N_2 to valves F^1 and *K*, by which first *K* and then *G* are thrown forward, reversing the main ports and completing the cycle. The speed of F^1 and *K*, and hence of *F* and *G*, is regulated by the screws *L*, L^1.

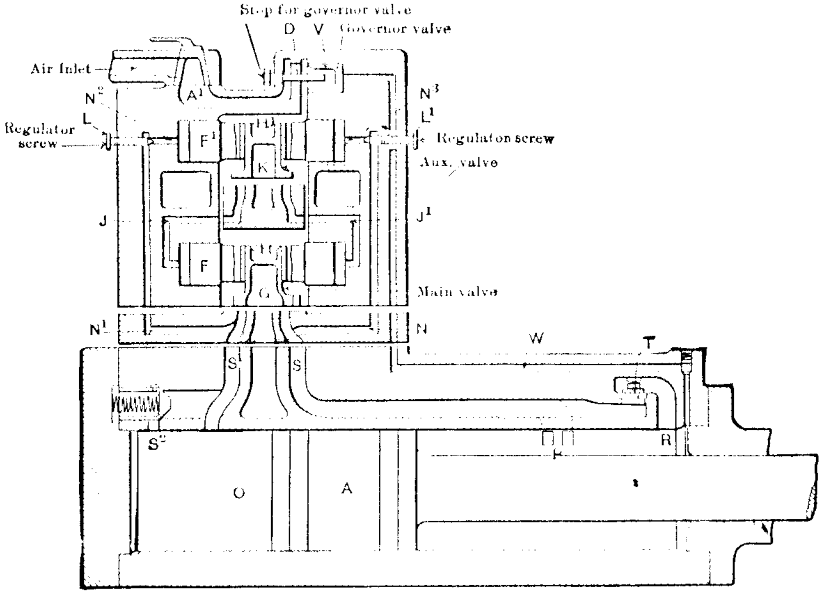

FIG. 14. Ingersoll-Rand Coal Pick. Diagram of Valves and Ports.

When the piston passes the double port *P* the exhaust ceases. Therefore, if the pick misses the coal, the piston on advancing beyond *P* cushions on air in the forward end of the cylinder. This high-pressure air forces back the governor valve *V*, wholly or partly cutting off the inlet air to valve *K*, so that the machine runs at reduced speed until the bit again strikes the coal before port *P* is covered. The regular exhaust then takes place, the governor valve *V* opens, and the machine resumes regular operation. The throw of *V* is adjusted as desired by the stop *D*. The back stroke is begun by the cushion air confined by the check-valve *T*. No live air can pass port *S* until the piston has advanced far enough to reduce the cylinder pressure below that of the live air in *S*, plus the resistance of the check-valve spring. The back stroke is cushioned by ports S^1 and valve S^2.

This pick is made in 3 sizes. Maximum depth of cut, 69 to 82 in.; weight, 675 to 850 lb. Standard wheels, 16 and 17 in.; for vertical, shearing cuts, 40 to 50 in.

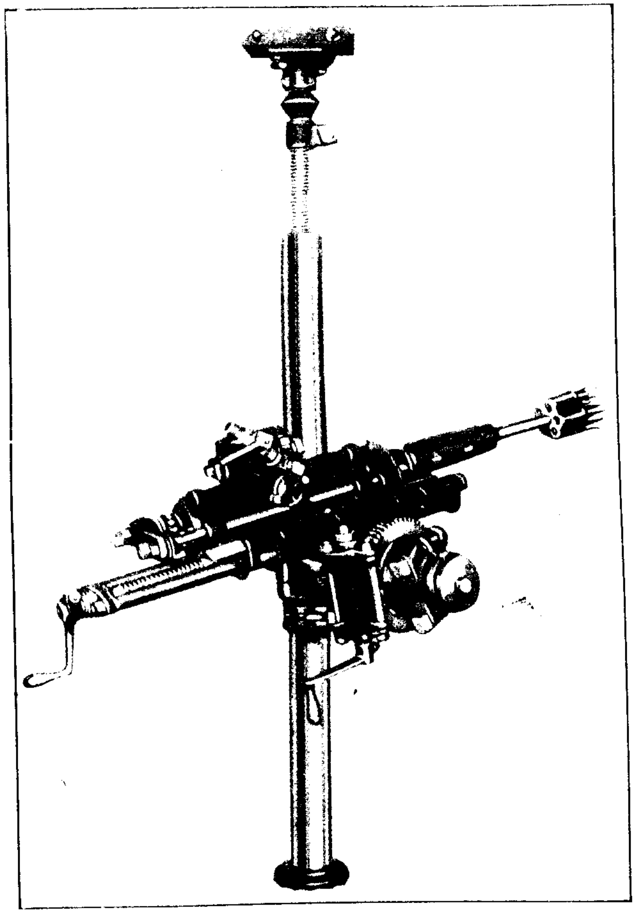

FIG. 15. "Radialaxe" Coal Cutter, Canadian Ingersoll-Rand Co.

Ingersoll-Rand Co. formerly made the "Radialaxe" coal cutter (Fig. 15) in the United States, but it is now made only by their Canadian branch. Used for undercutting in both longwall and room mining, and also for shearing, as in driving gangways, it is especially adapted to work in steeply pitching seams. A heavy drill cylinder (3$\frac{7}{8}$-in. diameter by 10-in. stroke) is mounted on a column and provided with a worm and worm-wheel sector. By this gearing the machine is swung in a horizontal or vertical arc, as required. Fig. 15 shows the set-up for shearing. The cut as first made is arc-shaped; then, by using successively longer steels, it finally attains uniform depth. For room-and-pillar mining, a single set-up usually suffices; for longwall, the column is shifted for successive cuts.

Instead of a single bit, a group of three or five chisel-edge bits is set in tapered sockets in a "rosette" bit holder. The latter is similarly socketed on a length of steel, which in turn is inserted in the chuck on the piston rod. The bits are readily removed for sharpening, being held by friction only. Extension steels of several lengths are provided for obtaining the necessary reach, as the cut is widened and deepened. From one set-up, the average depth of cut is 5½ to 6 ft., and the width, say 16 to 20 ft. Weight: unmounted machine, 218 lb.; 6-ft. column and worm gear, 186 lb.

Hardy Pick Co., Sheffield, England, makes a somewhat similar machine; weight of the entire outfit, 650 lb.; of the pick only, 250 lb. For longwall work, it may be mounted (with a short column) on a low truck, for convenience in shifting from one position to another.

Climax Rock Drill & Engineering Works., Carn Brea, England, also makes a column-mounted pick.

Unmounted Coal Picks. These, relatively light and inexpensive, have come into quite wide use during the past few years. In design they are essentially the same as the hand hammer tools known as pluggers, spaders and concrete breakers, except in such minor details as the chuck and the bit retainer. The bit is either pointed or chisel-shaped. Automatic rotation, so important for rock drilling, is unnecessary for the wide undercuts made in coal.

In the actual mining of coal, unmounted picks are used chiefly in two ways: (1) for making shallow undercuts of the shape shown in Fig. 12; (2) for increasing the height of the outer part of deep undercuts (kerfs) previously made by chain, bar or disk machines. The result of this operation, often called "snubbing," is that the block of coal so undercut, after being blasted loose from the roof, falls forward and breaks up for convenient loading into cars. As lighter charges of explosive can thus be used, less fine coal is produced. These picks serve also for cutting drains in gangways, "brushing," or taking down coal adhering to the roof, grading for tracks, breaking up large pieces of fallen roof rock, and taking up the bottom rock of workings in thin veins to increase their headroom.

Since these pick machines closely resemble the hammer drills of Class B (Chapter XIX) only a few brief statements are necessary here.

Chicago Pneumatic Tool Co., makes the pick shown in Fig. 16; a valveless machine, identical with their CP-3 spader, except in the shape of bit and chuck. The throttle trigger is operated by the palm of the hand. Net weight, 26–32 lb., depending on the type of handle; speed of stroke, 2,500 to 3,000 per minute.

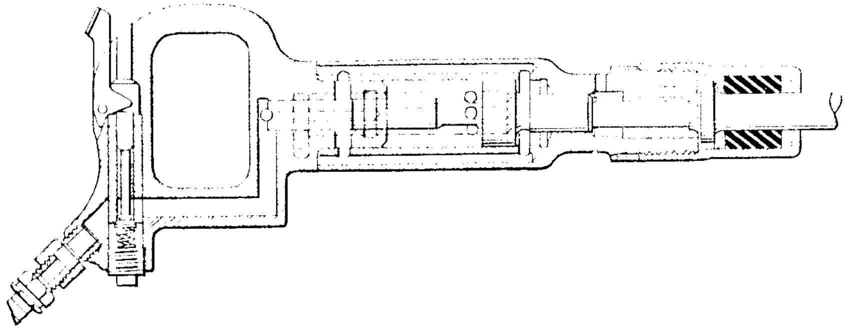

Fig. 16.—Coal Pick. Chicago Pneumatic Tool Co.

Consolidated Pneumatic Tool Co., Ltd., of Great Britain, a subsidiary of the Chicago Pneumatic Tool Co., has recently brought out the "Little Giant" pick. It is a light-weight, valveless tool, adapted both to coal mining and to such rock work as cleaning the face after blasting. Its chief uses in coal mining are those described above. Length of pick, 15 in.; blows per minute, 1,400; weight, 16½ lb.; free air consumption, at 80 lb., 25 cu. ft. per minute.

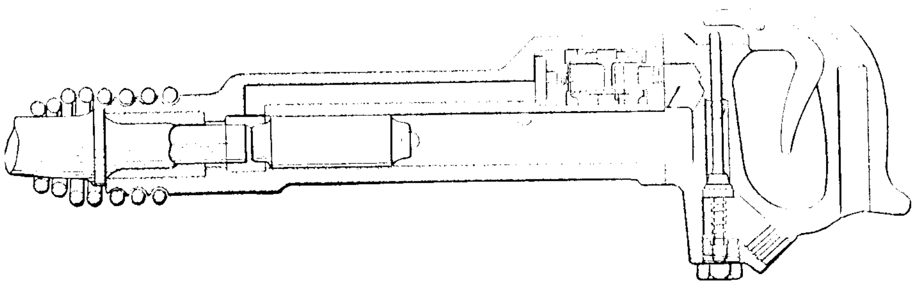

Fig. 17.—Coal Pick. Climax Rock Drill & Engineering Works.

Climax Rock Drill & Engineering Works, Carn Brea, Cornwall, England, makes a hand pick having a valve, shown by Fig. 17. Inside of the D-handle is a finger-operated throttle trigger. The weight is 25 lb. Other coal picks of this class are those of the **Ingersoll-Rand Co., Holman Bros., Sullivan Machinery Co.,** and **Dujardin et Cie.,** Lille, France.

Auger Drills, often used for boring holes for blasting coal, rock-salt and other soft material, are operated by compressed air or electricity, and are furnished by a number of makers. In the air drills, the auger is rotated through gearing, generally by a small, high-speed, 3-cylinder motor forming part of the machine. They are either held in the hands, like a breast drill, or mounted on a light column.

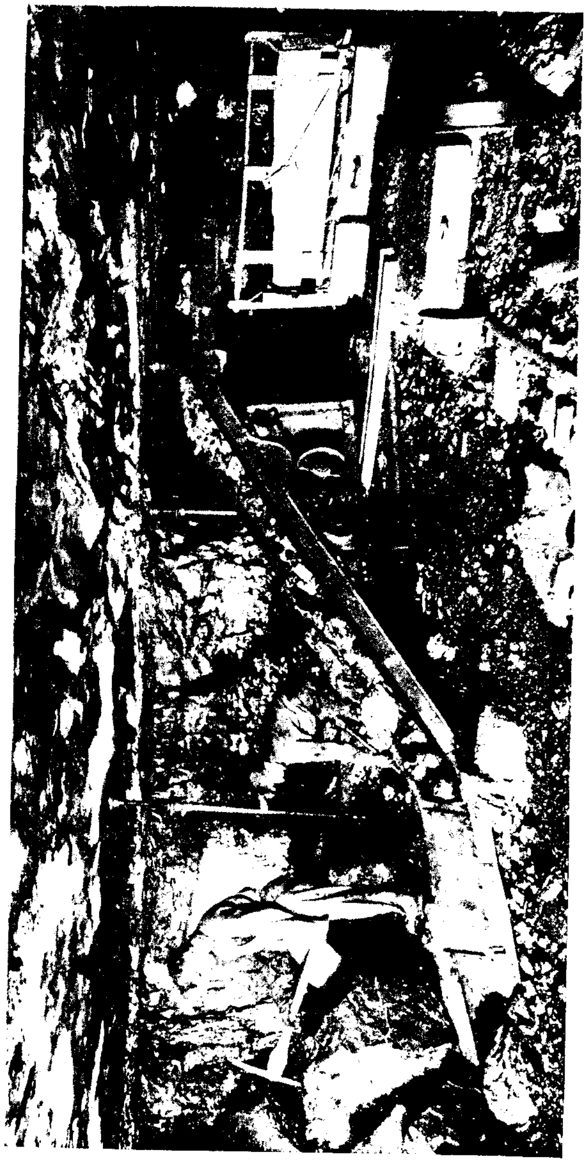

Fig. 18.—Jeffrey Air-driven Loader for Collieries.

Loading Machines, in a great variety of designs, are extensively used for loading cars in the working places of both coal and metal mines. They are now built by the Jeffrey Manufacturing Co., Sullivan Machinery Co. and others, with both electric and air-turbine drive; the latter being similar to that used for coal cutters (see *ante*, Figs. 3 5). Fig. 18 shows an air-driven loader of the conveyer type, for collieries. The frame of the engine supports the elevated end of the conveyer, near the car.

Quarrying Machinery

Quarrying is done (1) by channeling machines, which cut grooves or channels in the stone, and (2) by rock-drills, making rows of closely spaced holes for blasting or wedging.

Channelers are best adapted to the softer rocks, like limestone, marble, many sandstones, slate, and some varieties of granite and porphyry. Their special field of work is in getting out blocks of " dimension stone " for buildings; but they are often used in certain kinds of open-cut excavation, where smooth, uniform walls are desired, for example, rock cuttings for railroads, canals and water-wheel pits of hydroelectric plants.

Structural Characteristics of Channelers.—The machine consists essentially of a heavy cylinder, which, with its appurtenances somewhat resembles a large, reciprocating rock-drill, and which is mounted on a wheeled carriage. But, instead of having an ordinary bit, as for drilling holes, a " gang " of several square-shanked bits, set closely side by side, are clamped to the piston rod and cut a continuous, narrow groove or channel. The carriage stands on a 20-ft. length of track, laid alongside of the line of the channel to be cut, and is fed forward automatically, by a small single-cylinder, auxiliary engine and worm, geared by sprockets and chain to one of the carriage axles. Some channelers have a positive feed. Two of the truck wheels are then toothed, to engage with a rack which forms the rail on the side nearest the channel.

The motive power is steam, compressed air, or (rarely) electricity. A plant consisting of a few machines only may be advantageously run by steam; but, for large-scale work, especially when the machines operate at distances from one another, compressed air is more convenient and economical. The design of the channeler is essentially the same for steam as for compressed air, except that, with the latter, each machine should be provided with a reheater mounted on the carriage. Air pressures range from say 85 to 110 lb.

In most quarry work, the channels are cut at right angles, or nearly so, to the bedding planes of the rock. When the planes of stratification or cleavage are horizontal, the standard carrying the cylinder with its attached to the column arm encircles the feed cylinder, with a sliding fit. To hold the feed cylinder in position for the air pressure to react upon, a link of a chain pin-connected to the end of the cylinder is dropped into a groove on top of the clamp. For changing bits, the chain is simply lifted from the clamp, so that the entire machine can be run back from the drill hole.

Gardner-Denver Co. makes stoper drills in four models: M–11 and 77H are, respectively, light and heavy-duty machines, both provided with automatic rotation and designed for wet drilling only; 773 W and 116 VW are medium-weight, hand-rotated drills, and are designed in both dry and wet form. Fig. 65 shows the 77-H model. The design of the drill-head, including the valve action and rotation mechanism, is identical with that of the model 11 hand hammer drill (see Figs. 49 and 50). Model 116 VW is a valveless machine, the action of which resembles that of Model 16 VW (see Figs. 51 and 52).

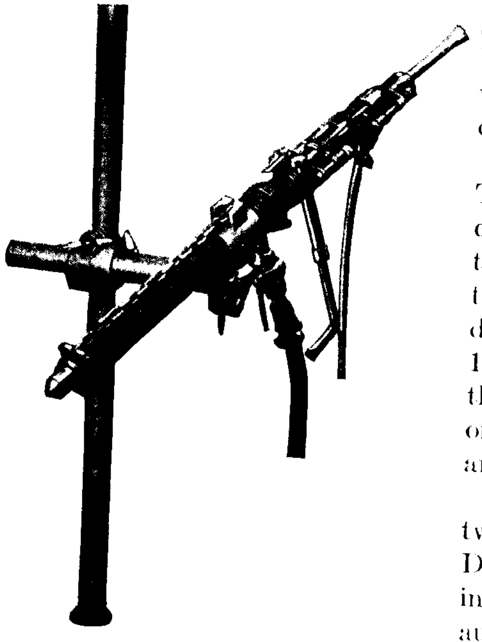

Fig. 64.—Holman Stoper Drill, Mounted.

Sullivan Machinery Co. makes two distinct types of stoper drill: DT-44, hand-rotated and furnished in both dry and wet forms; DU-48, automatically rotated and in wet form only. The DT-44 drill is valveless; the DU–48 has the same 3-ring spool valve as the Sullivan T-3 drifter drill (see Fig. 23 and accompanying description). For mounting on columns both drills are furnished with reverse feed.

Specifications of Sullivan Stoper Drills

Type	Bore, in.	Stroke, in.	Length overall, in.		Air hose, in.	Weight, lb.
			Feed closed	Extended		
DT-44	2¾	3	52	76	¾	{ Dry 72 { Wet 74
DU-48	2⅞	3	58¾	78¾	1	102

Note. Diameter of air-feed cylinder, 2¼ to 2½ in. For the wet drills, water hose is ¾ in. Front heads are designed for hexagon, quarter-octagon and cruciform steel; also for lugged shanks.

Chicago Pneumatic Tool Co.'s CP–146 stoper, made in both dry and wet forms, has the following specifications: bore, $2\frac{3}{4}$ in.; stroke, $3\frac{1}{2}$ in.; length overall, feed closed, dry 51 in., wet $52\frac{7}{8}$ in.; lengths extended, 75 and $76\frac{7}{8}$ in. respectively; weight, dry, 77 lb., wet, 82 lb. The valve is tubular, housed in renewable bushings in the wall of the cylinder and has the same design as that of the drifter drill CP-6 (see Fig. 25).

Stoper drills are also made by the following builders: Cleveland Rock Drill Co., Climax Rock Drill Co., Cochise Rock Drill Manufacturing Co., Gilman Manufacturing Co., Flottmann Machine Works, and Globe Rock Drill and Engineering Works. Further examples of Class C drills are unnecessary; their principal features have become standardized to a greater degree than the machines of Classes A and B.

Lubrication of Rock-drills.—This is an important factor in the maintenance of the machines and in securing good performance. The oil used must be heavier and more viscous than the ordinary "cylinder" and machine oils.

In general there are three methods: constant lubrication, "shot" lubrication, and that obtained from "line oilers." The first two methods have long been standard, and both are commonly applied in the same machine. The oil supply is contained in a small reservoir, attached to or integral with the drill-cylinder casting. In *constant lubrication*, the oil is fed automatically by the air pulsations produced by the piston strokes. In *shot lubrication*, a quantity of oil is admitted to the moving parts each time the air throttle is closed.

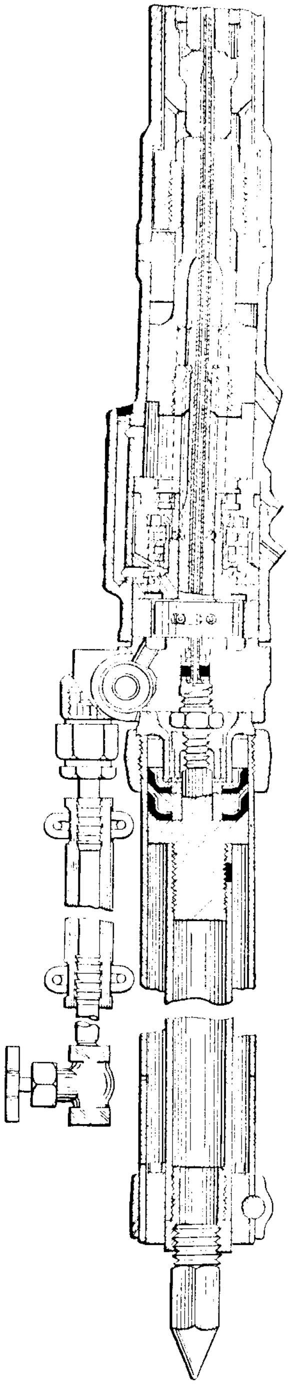

Fig. 65.—Gardner-Denver Stoper Drill 77-H.

Line oilers of different designs have recently been adopted by nearly all drill makers; the oil reservoir is placed in, and forms a part of, the air pipe or hose leading to the drill, the oil being siphoned from it by the compressed air and carried into all parts of the machine.

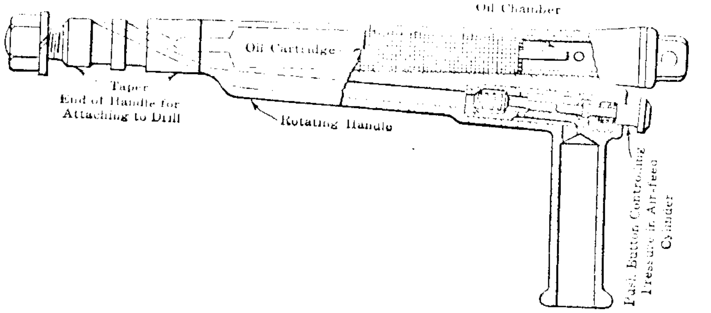
Oil Chamber
Oil Cartridge
Taper End of Handle for Attaching to Drill
Rotating Handle
Push Button Controlling Pressure in Air-feed Cylinder

Fig. 66.—"Heart-beat" Lubricator for Stoper Drills. Ingersoll-Rand Co.

Some large drifter drills have a special oiler in the valve chest, for lubricating the rear end of the cylinder; another oiler at the front end of the machine taking care of the chuck parts. The chuck is sometimes lubricated by medium heavy grease, charged once a shift into a chuck grease port.

An example of the devices for constant lubrication is the "heart-beat" oiler of the Ingersoll-Rand Co. In their drifter drills, this is placed in the valve chest and communicates with the cylinder through a small port; in the "Jackhamers," on the side of the cylinder (Fig. 37); and in the "Stopehamers," in the rotating handle of the drill (Fig. 66). It consists essentially of an oil-carrying cartridge of wire gauze, filled with absorbent material, and charged from the adjacent reservoir. In entering the drill valve and cylinder, the oil is atomized by the compressed air. The hollow drill handle containing the reservoir and cartridge fits in a taper hole in the valve chest, the oil channel being connected with the supply side of the valve.

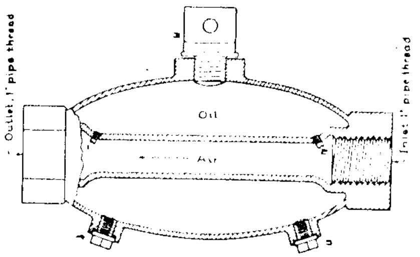

Fig. 67.—Air-line Oiler. Sullivan Machinery Co.

Air-line Oilers. Fig. 67 shows that of the Sullivan Machinery Co. The egg-shaped reservoir, ordinarily requiring two fillings per 8 hr., is inserted with nipples in the air hose, say 8 or 10 ft. from the drill. Referring to the illustration, *1* is the nozzle; *2*, vent; *3*, filler plug; *4* and *5*, cleaning plugs. Oils of different viscosities may be used by changing the diameter of the feed nozzle. Weight, 7 lb.

The Ingersoll-Rand Type D line oiler (brought out in 1928) is shown in Fig. 68. An annular reservoir *a*, holding one quart of oil, encircles the central tube *b*, which is connected up in the air-line to the drill. The reservoir has two filling holes, *c*, *c*, fitted with screw plugs. Coiled $1\frac{1}{2}$ turns around pipe *b* is a small tube *d*, in which there are two holes, *e*, *e*, covered by wire screens. When compressed air is turned on, oil is drawn from the reservoir into pipe *b* through the screened hole in tube *d* that happens to be below the oil level. In entering tube *d* with the air, the oil is atomized; this effect being aided by the pulsations of the air in the line-pipe when the drill is in operation. On stopping the drill, the flow of oil from the reservoir ceases. This oiler can not be taken apart; it is cleaned when necessary by removing the filling plugs and washing out with kerosene.

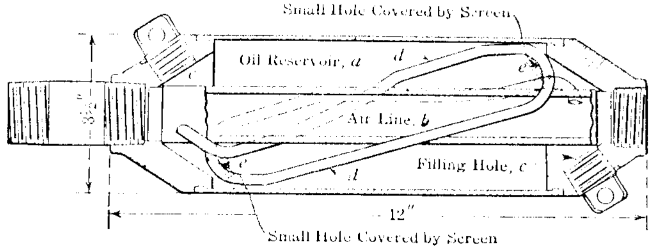

Fig. 68.—Ingersoll-Rand Type D Line Oiler.

The Gardner-Denver Co. makes two sizes of line oiler: the larger, weighing 11 lb. and holding 1 pint, is suitable for drifter and the heavier stoper and sinker drills; the smaller, weighing 6 lb. and of $\frac{1}{2}$ pint capacity, is for light hand-held drills, concrete breakers, etc. These oilers also operate on the siphon principle, are of the same shape as that in Fig. 67, but differ in details of design.

Makers of Rock-Drills.—The following list contains the names of most of the rock-drill manufacturers in the United States and Europe:

United States

Chicago Pneumatic Tool Co., Chicago, Ill.
Cleveland Rock Drill Co., Cleveland, Ohio.
Cochise Rock Drill Manufacturing Co., Los Angeles, Calif.
Gardner-Denver Co., Denver, Colo.
Gilman Manufacturing Co., East Boston, Mass.
Hardsocg Wonder Drill Co., Ottumwa, Iowa
Ingersoll-Rand Co., New York City
Sullivan Machinery Co., Chicago, Ill.

Great Britain

Armstrong, Whitworth & Co., Ltd., Newcastle-upon-Tyne
British Flottmann Drill Co., Ltd., Cardiff, Wales
Climax Rock Drill Co., Carn Brea, Cornwall
Consolidated Pneumatic Tool Co., Ltd., London
Globe Pneumatic Engineering Co., 1 Victoria St., London
Hardy Patent Pick Co., Ltd., Sheffield
Holman Brothers, Ltd., Camborne, Cornwall
International Channelling Machines, Ltd., Sheffield
Mining Engineering Co., Ltd., Worcester

Continent of Europe

Ateliers Liégeois d'Outillage Pneumatique, Ans-lez-Liège, Belgium
John Cockerill S. A., Seraing, Belgium
Duisburger Maschinenbau, Duisburg, Germany
Dujardin et Cie., Lille, France
Etablissements J. Massadier, Saint Étienne, France
Flottmann Aktiengesellschaft, Herne, Germany
Frölich u. Klüpfel, Unter-Barmen, Germany
Hauhinco Maschinenfabrik, Essen, Germany
Ruhrthaler Maschinenfabrik, Muhlheim-Ruhr, Germany

CHAPTER XX

PERFORMANCE OF ROCK-DRILLS

The work done by machines like steam- or air-driven engines and air compressors can be quite definitely determined, and their efficiencies can be expressed in terms that permit the making of useful comparisons between different machines. In Chapters II, X and XVI, rational expressions have been deduced for the performance of compressors and air motors.

Compressed-air drills, however, are in this respect in a class by themselves. It is well known that they are uneconomical in consumption of power, but it is difficult or impossible to make definite statements as to their efficiency. Their performance can not be determined absolutely in the ordinary mechanical sense. Although indicator cards can be taken, their significance is quite different from that of cards from a steam- or air-driven engine. Probably the best that can be done is to express the drill's work in empirical terms.

Work and Efficiency.—The useful work of a compressed-air drill is that which is expended in making a hole of certain depth and diameter in a rock of given hardness, toughness and physical character. To do this work, the drill requires a certain volume of air per minute at a given pressure, which has been compressed and delivered by the expenditure of an ascertained horsepower at the compressor. In other words, the drill's performance, expressed in terms of speed of drilling, or the number of linear inches of hole drilled per cubic foot of air consumed, is measured by the corresponding mechanical work done by the compressor; due allowance being made for the pipe transmission losses (see Chapter XV). This appears to be the most practical mode of estimating a drill's work and efficiency.

The rate of drilling or work accomplished, as expressed above, depends not only upon the physical and mineralogical character of the rock, together with the purely adventitious occurrence of seams and fissures, but also upon the mechanical condition of the machine and the personal element of the skill and experience of the drill-runner. The last-mentioned factor is of especial significance and can hardly be too strongly emphasized. A competent drill-runner will invariably do more work per shift than an inexperienced man; he will also make a faster advance in rock with which he is familiar than in rock that is new to him.

There is a further consideration respecting the work of rock-drills. Although the mechanical efficiency of engines in general is of prime importance, the efficiency of a drill must in practice be held subordinate to other attributes; ruggedness of construction, durability, simplicity of mechanism, portability and the facility with which repairs can be made. Drills must withstand hard and often unintelligent usage. Severe shocks and vibration are essential accompaniments of their operation. An outstanding characteristic of rock-drills is their ready applicability and suitability to their specific field of work.

In the operation of a piston drill, compressed air is admitted to the cylinder practically throughout full stroke. Expansive use of the air is practicable to a very limited extent, due to the variable length of stroke required. The drill, being mounted more or less rigidly on a tripod or column, can not be kept always at full stroke; it is often necessary to feed the drill-head so far forward that the length of stroke is shortened to one inch or even less, while the valve must still be able to cause a sharp, quick reversal. Useful work is evidently done only on the forward stroke, in striking the blow. If the valve is thrown too soon, the stroke will be shortened; if too late, the piston will strike the cylinder head. Rock-drills, therefore, can not attain the economy resulting in other air motors from using the air expansively (see Chapter XVI, Tables I and IV). Incidentally, admitting air at full stroke is of some advantage, because an exhaust at high pressure prevents troublesome accumulations of ice, in case the air is moist (see Chapter XVII).

Cushioning in Rock-drills.—With the idea of reducing shock, cushioning on the forward stroke was formerly a feature of some piston drills, but it is now recognized that for all types of drill an uncushioned blow is more effective. The maximum force of blow is thus attained, and the best drills are thus designed. The valve is not reversed until just before the blow of the bit is delivered. If, in drilling very hard rock, too heavy a blow is struck the remedy is to feed the drill-head forward, so as to work with shorter stroke.

On the back stroke, on the other hand, cushioning is desirable, to ease reversal of the piston and prevent possible injury due to its striking the cylinder head. The back-stroke cushion is produced by cutting off the exhaust before the end of the stroke. Only enough power needs to be developed on this stroke to overcome resistance of the moving parts and that caused by the frequent tendency of the bit to stick ("fitcher") in the drill hole.

It will be understood that the preceding statements apply only to piston drills, since their bits are rigidly attached to the piston rod and reciprocate with it. The bit of a hammer drill does not reciprocate with the hammer.

Speed of Stroke.—The speed of piston drills and of the large sizes of mounted hammer drills of Class A (see Chapter XIX) generally ranges from about 300 to 400 strokes per minute, the larger and heavier machines having the slower speeds. To give effect to the blow, the weight of the reciprocating parts must be relatively great; high speed tends to cause excessive wear and breakage. In the smaller hammer drills (including all those of Classes B and C), which strike a lighter blow at a high speed—say, 1,000 to 2,500 strokes per minute—the weight of the moving part (the hammer only) is comparatively small.

Air Pressure.—For piston drills, the best results in terms of work accomplished are obtained by using pressures of not less than about 85 lb., although in tunneling and similar work pressures up to 100 lb. are not uncommon. But high air pressures, especially when drilling in very hard rock, increase the repair cost, and the bits dull sooner and are more apt to chip. The relation between air pressure and speed of drilling was brought out clearly in the course of an extensive series of tests made at the gold mines of the Rand, South Africa, on piston drills operating under normal conditions. General average figures from these tests, on $3\frac{1}{4}$-in. piston drills, with 3-in. bits, are given in Table I. Readers desiring detailed data are referred to *Jour.* Transvaal Inst. of Mech. Engs., Nov., 1907, Feb., 1908, and *Eng. and Min. Jour.*, Jan. 21, Feb. 18, 1911.

TABLE I. EFFECT OF AIR PRESSURE, RECIPROCATING DRILLS

Six-minute runs	Gage pressure, lb.				
	80–70	70–60	60–50	50–40	40–35
Linear inches drilled per minute	1.3	1.1	1.0	0.6	0.5
Cubic feet free air per minute	124.	117.	100.	70.	60.
Cubic feet free air per linear inch of hole	95.3	106.4	100.	116.4	120.
Cubic feet free air per cubic inch of hole	13.3	14.8	13.8	15.0	16.6

For hammer drills the air pressure at the machine itself usually ranges between 70 and 100 lb.[*] As their average cylinder diameter is smaller than that of piston drills, the force of their blow is much less;

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[*] For results of an elaborate investigation of this subject at the United Verde copper mine, see *Trans.* Am. Inst. Min. Engs., Vol. 66, p. 743.

therefore, the considerations already set forth as to the effect of high air pressure on drill maintenance do not apply equally to these machines.

Consumption of Air and Compressor Capacity.—The duty of a machine drill is usually based on its consumption of free air per minute, which depends on the cylinder diameter, speed of stroke, air pressure, the proportion of the total time actually occupied in drilling, and character of the rock. In discussing this important but rather difficult subject, certain relations between the actual air consumption by the individual drills and the required capacity of the compressor must be kept in mind.

The compressor capacity for one drill is evidently greater than the average required for a number of machines. With a large number, the delays to which each is subject, for setting up or shifting, changing bits, stoppages caused by sticking of the bit in the hole, etc., make it improbable that all will be in operation at any one time; hence, the average allowance of air for each may be reduced. Occasional load peaks of short duration will be cared for by the air receiver storage.

Table II, showing the free air per minute required for one drill of the piston (reciprocating) type, is based on a comparison of the statements of several manufacturers, checked by recorded tests. It represents, within reasonable limits, actual practice for these machines when in good order. This table, although primarily for piston drills, is roughly applicable also to hammer drills of the same cylinder diameter.

TABLE II.—CUBIC FEET OF FREE AIR PER MINUTE CONSUMED BY ONE DRILL AT SEA-LEVEL

Gage pressure	Diameter of Drill Cylinder in Inches											
	2	2¼	2½	2¾	3	3⅛	3 3/16	3¼	3½	3¾	4¼	5
60	58	63	70	82	90	97	100	105	111	118	135	155
70	62	72	80	92	104	112	115	118	130	135	152	171
80	70	80	88	103	115	125	130	135	142	153	173	205
90	78	87	95	115	128	137	144	148	165	173	191	222
100	85	96	108	126	140	151	155	161	176	184	210	250

When a number of drills are operated by the same plant, since it may be assumed that not all of them will be running at any one time, their total average air consumption may be estimated by applying the multipliers in Table III. To find the average air consumption (or the compressor capacity required), multiply the cubic feet of free air per minute for one drill (Table II) by the multiplier corresponding to the total number of drills operated (Table III).

TABLE III.—MULTIPLIERS FOR ESTIMATING THE AIR CONSUMED BY A NUMBER OF DRILLS

Number of drills	1	2	3	4	5	6	7	8	9	10
Multiplier	1	1.8	2.7	3.4	4.1	4.8	5.45	6.1	6.7	7.3
Number of drills	11	12	15	20	25	30	35	40	50	60
Multiplier	7.8	8.4	10.3	12.8	15.1	17.3	19.7	22.0	26.5	30.5

As already stated, Table III is approximately correct for both piston and hammer drills. It is true that hammer drills have a much higher speed of stroke, and therefore consume more air per unit of actual drilling time. But, under nearly all conditions, they drill faster, in terms of linear inches of hole, than piston drills of the same cylinder diameter. Therefore, for a given amount of work, smaller hammer drills can be used.

For further aid in making estimates, Table IV may be found useful.

TABLE IV.—AIR CONSUMPTION OF HAMMER DRILLS (TESTS AT UNITED VERDE MINE)

Type and size of drill (all "wet")	Air pressure, lb.							
	50	60	70	80	90	100	110	120
	Cu.ft. of free air per min.							
Mounted:								
Leyner-Ingersoll, 3-in.	74	95	113	133	150	170	189	209
Leyner-Ingersoll, 2¼-in.	72	80	90	105	115	128	140	153
Leyner-Ingersoll, 2½-in.	75	85	100	112	125	140	148	156
Sullivan DR-6, 2½-in.	77	96	115	132	150	164	180	193
Sullivan DX-61, 3-in.	86	105	125	145	165	184	203	220
Waugh No. 66, 3-in.	88	112	134	156	180	202	223	240
Waugh No. 60, 2¾-in.	77	92	105	120	137	148	168	182
Jackhammers:								
Ingersoll-Rand BCR-430, 2¼-in.	47	56	64	73	80	91	106	108
Sullivan DP-33, 2¼-in.	56	66	77	88	100	110	121	130
Waugh 21V, 2½-in.	62	75	85	95	107	120	133	144
Stopers:								
Ingersoll-Rand CC-14, 2¼-in.	53	65	76	84	92	101	111	120
Sullivan DT-44, 2¼-in.	70	85	100	116	130	143	155	165

It contains a portion of the figures on air consumption obtained from 1,500 tests, made on the hammer drills specified, at the United Verde copper mine, Arizona. Details of these tests are given in *Trans. Am. Inst. Min. Engs.*, Vol. 66, p. 743. All the drilling was done in massive sulphide ore of uniform hardness. A "Drillometer" (Chapter XXV) was used for measuring the air consumption. In making these tests, the only variable for each drill was the air pressure, with corresponding changes in free air consumption and in the number and force of the blows. It will be noted that the air consumption, under the specific conditions named, ranges somewhat higher than the figures in Table II.

Factors in Drill Operation.—It will be understood that the figures given in the preceding tables are not exactly applicable to all cases. Modifying factors are:

(1) *Type of Drill.* Since piston drills are mounted on tripods or columns, and their bits are clamped solidly in the chuck by bolts, it takes much longer to set them in position for drilling and to change bits than is the case with hammer drills.

(2) *Kind of Work.* The time required to set up a piston drill also depends on the shape of the working, whether a tunnel or drift, a shaft, stope, or open cut. If the floor and roof, or the side walls, of a mine opening are irregular or loose, much time may be needed to shift the machine and set it up, especially if it is mounted on a column.

(3) *Character of Rock.*—In hard rock the advance in drilling is slower than in soft, so that the machine makes longer continuous runs. Less time is occupied in shifting and setting up for drilling the successive holes of a round, and the air consumption per unit of time is therefore greater. Although this increase is partly offset by the fact that the bits are more quickly dulled in hard rock and must be changed at shorter intervals; still, in very hard ground the drills may be kept running with but few and short intermissions. In soft rock, though the actual speed of drilling is greater, there are apt to be more delays due to rifling of the hole and sticking or "fitchering" of the bit. The time actually occupied in drilling will vary for each piston drill from, say, 4 to 6 hours out of an 8-hour shift; for each hammer drill, say 6½ to 7 hours per shift.

(4) *Physical Condition of the Drill.*—The figures in Tables II and III are for machines in good condition. More air is consumed by old drills, with valves and pistons so worn that they do not fit closely. Their exhaust, instead of being short and sharp, is often nearly continuous. Repairs of drills are dealt with later.

This subject merits closer attention than it usually receives at plants where no check on air consumption is maintained. When compressed air is furnished by large central plants at meter rates, as at some of the groups of gold mines on the Rand (see Chapter XXV), efficient maintenance of the drills is stimulated, and pipe leaks are more carefully looked after, because the exact air consumption at a given mine, or for certain groups of drills, can be accurately known. For suggestions as to the use and care of drills see *Eng. and Min. Jour.*, Sept. 8, 1923, p. 423.

To Find the Compressor Horsepower for any number of drills at any altitude.

Example.—10 2¼-in. drills, working at 12,000 ft. altitude, with air at 80 lb. gage.

From Table II, 1 drill at sea-level uses 80 cu.ft. free air, and 10 drills (Table III), $80 \times 7.3 = 584$ cu.ft. At 12,000 ft., as the relative output for 80 lb. (Chapter XII, Table I) is 0.65, the compressor capacity is $584 \div 0.65 = 900$ cu.ft. The volume of compressed air per indicated horsepower at 80 lb. and 12,000 ft. $= \frac{0.925 + 0.646}{2} = 0.785$ cu.ft. In the equation $V' = \frac{VP}{P'}$, P, from Chapter XII, Table I, $= 9.34$; hence, $V' = \frac{900 \times 9.34}{80 + 9.34} = 94$, and $\frac{94}{0.785} = 119.7$ hp.

The horsepower may also be found by using the ratio of compression directly, thus: cu.ft. at sea-level (see above) = 584. Ratio of compression at 12,000 ft. $= \frac{80 + 9.34}{9.34} = 9.56$, which (Chapter X, Table III) corresponds to 127 lb. gage, and the horsepower required to compress 1 cu.ft. to this pressure = 0.204. Hence, $584 \times 0.204 = 119.13$ hp., which agrees closely with the result obtained by the first method.

Drilling Records.—In recording the work of machine drills, the principal items are:

(1) Kind of rock or ore; (2) type and size of drill; (3) air pressure at the drill; (4) diameter of hole; (5) total elapsed time for a hole of given depth; (6) time occupied in setting up, changing bits, and delays due to breakage or derangement of the drill, or to "fitchering" of bits; (7) net drilling time per hole and per foot or inch of hole, resulting in the number of inches per minute (or feet per hour) drilled while the machine is in actual operation.

If approximately accurate values are secured for each of these factors, the comparative speeds of drilling in different rocks and ores could be tabulated with advantage. But, because of the almost infinite variations in rock and ore characteristics, and other local conditions, detailed comparisons are of doubtful value. The results of "test runs" are sometimes cited to show that one machine is better than another, in the sense that it can drill faster or that it uses less air for the same footage of hole. By operating drills of different type side by side, in the same rock and with the same air pressure, some approach to an accurate comparison would be possible; but, even then, it is clear that the physical condition of the competing drills, and the "personal equations" of the respective drill runners, affect the rate of work and cost per foot of hole. Moreover, as test runs are made under the stimulus of rivalry (and perhaps with a "bonus" to the winner), they do not constitute a sound basis for ascertaining the relative merits of different drills, nor the footage that could reasonably be expected in ordinary daily work. For further elucidation of these considerations the reader is referred to the voluminous records of the elaborate drill tests made on the Rand (see p. 383).

It must be concluded that the most practical and rational mode of dealing with this rather difficult subject is to make a time study of each individual case.

Time Studies of Rock-drill Work.—Instead of using Tables II, III, and IV for estimating the compressor capacity required for a certain number of drills, the numerical values of the factors involved in a drill's operation may be ascertained. Since the effects of the actual local conditions in any given case are thus determined, more accurate results are secured than those obtained from tables, which are necessarily based upon assumed or general conditions. The following analysis is taken from "Rock Drilling," p. 58, by Dana and Saunders.

Time to change bits and clean the hole, e minutes; to drill 1 ft., d minutes per foot; to move drill and lost time, l minutes; to set up drill, g minutes.

Length of feed, feet, f; depth of hole, feet, D; time to drill length of feed (see note p. 389), $fd = r$. Number of bits per hole, $D \div f$.

Total time per hole $= (e + fd)(D \div f) + (l + g)$, including moving and setting up drill. Working minutes per day $= M$ (say, 600).

$$\text{Number of holes per day} = \frac{M}{(e + fd)(D \div f) + (l + g)}$$

$$\text{Feet of hole drilled per day} = \frac{DM}{(e + fd)(D \div f) + (l + g)}$$

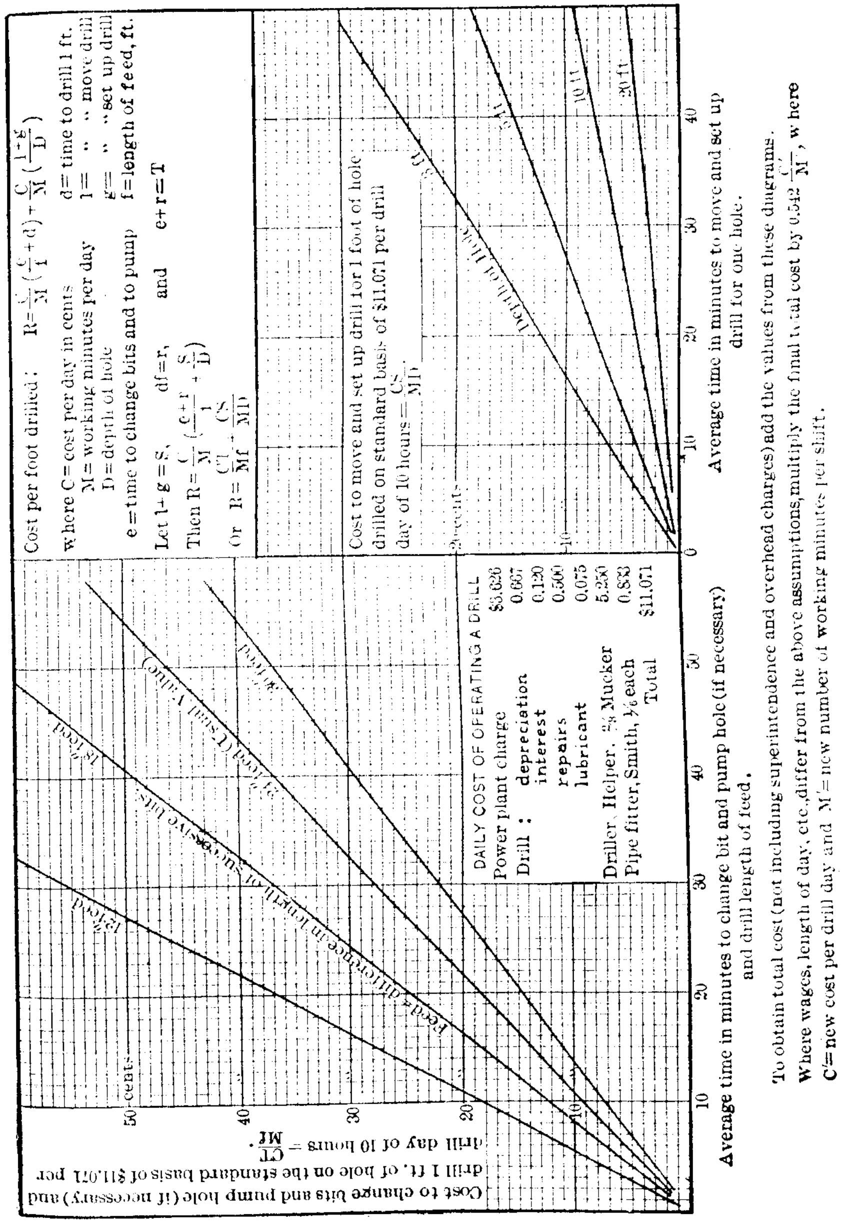
Cost per day $= C$ on standard basis: hence, cost per foot of hole

$$= \frac{C(e + fd)(D \div f) + (l + g)}{DM} = R = \frac{C}{M}\left[\frac{e}{f} + d + \frac{l + g}{D}\right]$$

If $l + g = S$, or average time to move drill and set up; and $e + r = T$, or average time for changing bits, cleaning hole, and drilling the length of feed, the above formula reduces to:

$$R = \frac{CT}{Mf} + \frac{CS}{MD}$$

Note that the length of feed (f) is the difference in length between successive bits, which is ordinarily from $1\frac{1}{2}$ to 2 ft. In the formula the maximum feed for which the drill is designed is immaterial; the value is affected only by the lengths of bit. If it be difficult to get a follower bit



Cost per foot drilled: $R = \frac{C}{M}\left(\frac{e}{f} + d\right) + \frac{C}{M}\left(\frac{l + g}{D}\right)$

where C = cost per day in cents
M = working minutes per day
D = depth of hole
e = time to change bits and to pump
d = time to drill 1 ft.
l = " " move drill
g = " " set up drill
f = length of feed, ft.

Let $l + g = S$, $df = r$, and $e + r = T$

Then $R = \frac{C}{M}\left(\frac{e + r}{f} + \frac{S}{D}\right)$

Or $R = \frac{CT}{Mf} + \frac{CS}{MD}$

DAILY COST OF OPERATING A DRILL		
Power plant charge		$3.626
Drill:	depreciation	0.667
	interest	0.120
	repairs	0.500
	lubricant	0.075
Driller, Helper, ⅞ Mucker		5.250
Pipe fitter, Smith, ⅙ each		0.833
	Total	$11.071

To obtain total cost (not including superintendence and overhead charges) add the values from these diagrams.

Where wages, length of day, etc., differ from the above assumptions, multiply the final total cost by $0.542\frac{C'}{M'}$, where C' = new cost per drill day and M' = new number of working minutes per shift.

COST OF DRILLING WITH COMPRESSED AIR

into the hole, the feed of the drill should be several inches longer than the difference between lengths of bits, since this affects the time of changing bits and cleaning the hole.

For any given case, the values of the preceding factors can be readily obtained with a watch and note book, within small limits of error. As a further aid, the estimated costs found from the analysis are platted in the diagram on p. 389, from which can be read directly the cost per foot of hole drilled.

Examples of Drilling Speeds based on recorded work under a variety of conditions, are given in Tables V and VI. They will aid in making rough estimates. See also "Handbook of Rock Excavation," by H. P. Gillette.

The average speed of piston drills based on column 6 of Table V is 6.5 ft. per hour. In general the duty of a standard 3-in. drill, in rock or ore of average hardness, ranges from 40 to 50 ft. per 8-hour shift, including all ordinary delays for setting up and changing bits. For very hard, tough ground, the speed is often lower, while much more than 50 ft. per shift may be made when the conditions are favorable, and also in drilling deep holes, for which fewer set-ups are required. For moderately soft ground, not requiring holes of large diameter to contain the necessary quantity of powder, the smaller sizes of machine drill (2–$2\frac{1}{2}$ in.) are usually preferable. Their first cost and air consumption are less than for large drills, and they can be operated by one man. Small machines are especially useful for stoping in thin veins. For hard ground, and as a rule in shaft sinking, tunneling, cross-cutting and similar work, the $2\frac{3}{4}$-, 3-, and $3\frac{1}{4}$-in. sizes are best. For deep holes in open-cut work, still heavier drills are often necessary — up to $3\frac{1}{2}$ in., or even larger.

The average speed of hammer drills, based on the figures in column 5 of Table VI, is 12.6 ft. per hour of total elapsed time.* Their faster work, as compared with that of piston drills, is due mainly to three causes: (1) machines of Classes B and C, being generally unmounted, are set in position for work with little or no loss of time; (2) since there are no chuck bolts to manipulate, the bits of all three classes of drill are changed quickly (in say $1\frac{1}{2}$ to 2 minutes); (3) hollow steel, for the passage of compressed air or water, being nearly always used, the bottom of the hole is kept constantly clear of pasty sludge, thus permitting the hammer to make more effective blows on the bit.

As a rule, the fastest work is done when drilling "uppers" (holes directed at a steep upward angle), as in stoping. When the ore or rock is dry, the cuttings will then often run out by gravity.

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* For additional data, see *Trans.* Am. Inst. Min. Eng., Vol. 51, Table on p. 245; Vol. 66, p. 717.

TABLE V.—DRILLING SPEEDS, PISTON DRILLS

Kind of Work	Rock or Ore	Size of drill, in.	Air pressure, lb.	Average depth of hole, ft.	Inches of Hole per Minute	
					Total time	Net time
Crosscut, 5 × 7 ft.	Hard quartzite and limestone.	3	60	3	0.43	
Crosscut, 5 × 7 ft.	Hard quartzite and limestone.	3	60	$3\frac{3}{4}$	0.40	
Drift	Quartzite	$3\frac{1}{8}$	80	$5\frac{1}{2}$	1.29	1.55
Tunnel, 7 × 7 ft.	Basalt	3	......	......	0.72	
Stoping	Amygdaloid copper rock.	$3\frac{1}{8}$	63	7	0.93	
Stoping	Amygdaloid copper rock.	$3\frac{1}{8}$	63	6	1.60	2.30
Drift	Amygdaloid copper rock.	$3\frac{1}{8}$	65	3	1.26	2.70
Stoping and drifting	Hard limestone	$2\frac{3}{4}$ $2\frac{1}{4}$	80	7	1.00	
Stoping	Hard limestone	$2\frac{1}{4}$	75	$6\frac{3}{4}$	......	2.46
Stoping	Magnetic iron ore	3	80	$6\frac{1}{2}$	1.31	
Stoping	Rather soft porphyry, etc.	$2\frac{3}{4}$	75	$4\frac{2}{3}$	1.38	
Stoping	Very hard quartzite	$2\frac{3}{4}$	75	$3\frac{1}{5}$	0.54	
Crosscut	Quartz and hard slate	$2\frac{3}{4}$	73	......	1.50	
Stoping	Hard quartz	$2\frac{1}{4}$	73	......	1.20	
Stoping	Hard hematite	3	74	7	1.28	
Stoping	Quartzose	$2\frac{3}{4}$	......	$5\frac{1}{3}$	1.51	2.25
Drift, 10 × 10 ft.	Hard limestone	$3\frac{1}{4}$	......	4	1.35	
Stoping	Pyritic ore	$3\frac{1}{8}$	......	4	1.20	
Drift, 4.5 × 6 ft.	Limestone	$2\frac{3}{4}$	105	$8\frac{1}{4}$	2.62	
Stoping	Amygdaloid copper rock.	3 & $3\frac{1}{8}$	......	7	1.67	2.45
Drift	Amygdaloid copper rock.	3 & $3\frac{1}{8}$	......	$5\frac{2}{3}$	1.46	2.34
Drift	Amygdaloid copper rock.	3 & $3\frac{1}{8}$	......	$5\frac{2}{3}$	1.61	1.97
Stoping	Hard phonolite breccia	$2\frac{1}{4}$	......	$2\frac{1}{3}$	0.89	0.95
Drift	Hard phonolite breccia	$2\frac{1}{4}$	......	$4\frac{1}{2}$	2.60	3.38
Crosscut, 9 × 9 ft.	Quartzite	$2\frac{3}{4}$	......	$3\frac{2}{3}$	1.17	2.43

Field of Work of Hammer Drills.—Class A drills are best adapted to tunneling or other horizontal work. Class B machines are especially designed for "down holes," as in shaft sinking, quarrying, trenching, and

TABLE VI.—DRILLING SPEEDS, HAMMER DRILLS

Kind of Work	Rock or Ore	Type of drill, see Chapter XIX	Depth of hole, ft.	Inches of Hole per Minute	
				Total time	Net time
Drifting	Limestone	Class A	......	3.10	
Stoping	Hard phonolite breccia	Class C	2.2	2.04	2.54
Drifting	Tough schist	Class A	......	2.64	
Stoping	Hard trachyte	Class C	4.0	2.35	2.90
Stoping	Hard porphyry	Class C	5.5	2.09	
Drifting	Amygdaloid copper rock	Class A	6.0	2.15	3.22
Stoping	Quartzose ore	Class C	3.0	2.15	3.58
Drifting	Granite	Class A	3.0	3.00	
Drifting	Quartzose ore	Class A	3.3	1.58	
Stoping	Medium andesite	Class C	4.3	2.85	3.52
Stoping	Pyritic ore, medium hard	Class C	5.3	2.00	
Tunneling	Hard granite	Class A	7.3	4.82	7.16
Tunneling (test run)	Average granite	Class A	7.0	3.56	5.16
Tunneling	Hornblende schist	Class A	6.5	2.64	
Sinking	Tough schist	Class B	6.5	1.43	
Drifting	Hard limestone	Class A	3.2	1.63	1.86
Raising	Quartzose ore	Class C	4.0	2.40	5.10
Tunneling	Granite	Class A	3.1	3.00	

general surface excavation of rock. Class C drills are intended for, and do their best work in, drilling "uppers," as in overhand stoping. They are useful, also in mining thin veins with narrow paystreaks. For breast holes, they are occasionally mounted on columns or tripods.

TABLE VIa.—ROCK-DRILL PRICES (1930)

Type	Weight, pounds	Range of prices
Hand-hammer drills	28–72	$140–$270
Sinker drills	75–113	290–325
Drifters	117–269	335–550*
Stopers:		
Hand rotation	63–105	205–260
Automatic rotation	89–139	260–380
Paving breakers	56–79	165–195

* This price includes extra long feed-shell for deep holes.

Cost of Drills.—The cost of drills varies somewhat among the different makers, but Table VIa shows the general range of prices of drills of various classes and types. These figures will be of use at times, in making preliminary estimates of cost of plant.

Drill Repairs.—This is a subject of importance.

The work of rock-drills is exceptionally severe. Even when they are operated with care, wear is rapid and derangement or breakage frequent. The item of depreciation is large. The cost of drill maintenance depends chiefly upon the following considerations:

(*a*) The design of the drill and grade of the materials entering into its construction. The maker has the problem of producing a machine having the minimum number of individual parts, especially moving parts, consistent with a good mechanical design. Only the best materials are admissible. Parts particularly liable to wear or breakage should be replaceable readily and inexpensively, and, so far as possible, without rendering it necessary to discard a larger piece with which the broken part may be solidly connected.

(*b*) Care and experience are indispensable in selecting the type and make of drill best adapted to the service required.* To facilitate repairs, it is advisable to use at any given mine or rock excavation as few different makes and sizes of drill as practicable. Observance of this general rule will evidently reduce the number of spare or duplicate parts that must be kept in stock.

(*c*) The experience and character of the drill-runner. A competent, careful man treats his machine with intelligent consideration. When operating a mounted drill, he will set it up properly, to minimize the risk of its getting out of alinement as the hole is deepened; and, if the bit should stick ("fitcher") he keeps his temper and refrains from striking unnecessarily heavy blows on the drill-head or chuck. A fitchered bit can often be loosened by slacking the clamp bolts by which the drill-head is fastened to the mounting, thus allowing the machine to shift its position slightly with respect to the axis of the hole. The abuse to which machine drills are frequently subjected may be reduced by efficient supervision on the part of foremen and shift-bosses.

(*d*) Maintenance of an adequate repair force and equipment. A competent master mechanic can do much, not only to minimize the direct cost of repairs, but also, by keeping all the drills in good condition, to prevent incidental losses due to cessation of work. It must be remembered that, when a drill has to be stopped for repairs, the effective work of one or more men is interrupted. As a general rule, drill-runners should not be encouraged to tinker their machines underground. If serious repairs or adjustment are necessary, the drill should be sent at once to the shop.†

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* See *Engineering and Mining Journal*, Oct. 30, 1915, p. 719.

† For further discussion of maintenance and repairs see Chapter VIII, "Rock Drills," by E. M. Weston. See also *Mining and Scientific Press*, 1905, Nov. 4, p. 308; Nov. 11, p. 329, and *Engineering and Mining Journal*, Nov. 11, 1916, p. 861, Sept. 8, 1923, p. 423.

When new, the piston of a drill should have the closest possible working fit; the difference between the cylinder bore and piston diameter not exceeding $\frac{1}{64}$ in. Modern drill cylinders are counterbored at each end, to facilitate reboring. New cylinders may be bored to fit worn pistons, or new pistons fitted to old cylinders. The difference in diameter between a spool-valve and the bore of its chest should not exceed say $\frac{1}{100}$ in. It is important to keep the chuck and chuck keys in good order, and the bit when clamped must be in true alinement with the piston.

Table VII. Repair Costs of 15 New $2\frac{1}{4}$-Inch Drills

Drill	Days in Use, Two Shifts per Day						
	25	50	75	100	125	150	175
1	$0.60	$1.25	$1.50	$2.40	$3.00	$7.05	$10.80
2	3.75	7.50	7.90	8.75	12.00	16.25	21.80
3	3.30	10.00	11.10	13.50	20.30	40.75	40.91
4	.90	1.25	4.55	5.30	28.39	28.39	28.39
5	.15	.40	1.50	4.50	5.80	6.40	9.20
6	3.80	7.40	8.00	8.08	8.08	12.15	
7	.20	.50	1.10	4.75	8.00	14.95	16.20
8	.55	1.05	5.60	5.91	7.80	10.57	10.70
9	4.00	6.20	8.70	12.30	41.20	54.59	
10	.70	70	3.50	4.70	8.29	8.45	
11	.60	1.00	1.50	21.00	41.00	42.10	43.36
12	.55	1.00	2.20	9.10	12.00	12.30	21.30
13	.85	1.20	4.40	7.70	8.70	10.25	18.14
14	3.65	1.10	4.85	5.05	8.50	50.50	52.01
15	2.55	4.35	4.65	5.20	11.20	14.60	14.75
Totals	$26.15	$17.90	$78.05	$118.21	$221.26	$329.30	
Averages	1.74	3.19	4.74	7.88	11.95	21.95	

Repair Costs. While no generalizations are possible, on account of the extremely variable service of rock-drills, Tables VII and VIII, by C. K. Hitchcock, Jr. (*Eng. and Min. Jour.*, May 9, 1914), of repair costs of piston drills at a Michigan copper mine, will be found useful.* The records of cost of labor and material (not including mountings and connections) were kept by the month, the cost at the end of each month being platted on a chart against the number of days, of two shifts, that the drill had worked. From the chart the costs for each period of 25

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* For maintenance costs at Rand mines, see *Engineering and Mining Journal*, Jan. 11, 1914, p. 223.

Table VIII.—Repair Costs of Six $3\frac{1}{8}$-Inch Piston Drills

Drill	Number of Days in Use					
	25	50	75	100	125	150
A	$2.00	$2.50	$4.00	$5.10	$8.00	$19.00
B	3.00	11.30	19.10	22.30	27.00	31.00
C	6.10	14.20	18.20	19.60	24.50	28.60
D	4.20	5.20	6.20	9.20	18.10	18.20
E	28.00	33.30	36.10	37.00	38.10	41.50
F	50	4.30	4.90	11.00	12.80	14.00
Totals	$43.80	$70.80	$88.50	$104.50	$128.50	$155.30
Averages	7.30	11.80	14.75	17.42	21.42	25.88

Drill	Number of Days in Use					
	175	200	225	250	275	300
A	$22.00	$22.60	$40.00	$64.20	$84.80	$88.10
B	40.20	41.30	42.20	46.70	53.20	59.60
C	28.90	28.90	32.60	33.40	46.30	49.20
D	18.30	21.50	22.10	46.00	46.00	47.60
E	44.70	45.90	46.50	46.90	48.20	49.50
F	17.80	18.20	18.20	35.00	55.20	59.30
Totals	$171.90	$178.40	$201.60	$272.20	$333.70	$353.30
Averages	28.65	29.73	33.60	45.37	55.62	58.88

Drill	Number of Days in Use					
	325	350	375	400	425	450
A	$97.00	$99.50	$101.50	$102.00	$115.20	
B	66.00	66.80	67.00	78.80	79.50	
C	50.20	48.80	54.50	61.00	75.00	$76.70
D	48.20	53.70	54.60	57.10	60.20	62.70
E	72.00	90.50	111.20	112.00	113.80	113.80
F	60.20	60.70	63.00	63.70	64.00	70.00
Totals	$393.60	$420.00	$451.80	$474.60	$508.00	
Averages	65.60	70.00	75.30	79.10	84.67	

days were scaled (see the tables). Most of the smaller repair parts were made in the mine shops, the larger parts being bought from the makers. Table VII shows that the average costs between 100 and 150 days of service were about $7.00 for each 25 days.

Table VIII indicates a repair cost of about $5.00 per 25 days. In the upkeep of machine drills, both economy and efficiency can be improved by discarding drills having abnormally poor records, or by overhauling them thoroughly, to reduce cost of maintenance.

At a large group of mines on the Rand, the complete repair cost of standard piston drills (including air hose) averaged $42.00 per 52 shifts. On introducing a contract system of drill repairs, the cost was reduced to $27.00, and it was thought it might be further reduced to $21.00 or $23.00.

Table IX contains examples of drill repair costs in tunnel work. In excavating the small cross-sections of tunnels these costs are always higher than when the holes are drilled in the larger faces of mine stopes, because the amount of rock broken per foot of hole is relatively small.

TABLE IX.—COST OF DRILL REPAIR PARTS*
(D. W. Brunton and John A. Davis)

Tunnel	Year	Feet driven	Rock	Drill	Cost per foot of hole, cents
Carter	1911	3,000	Granite	Leyner	2.0
Gunnison	1906	......	Granite and sandstone	Sullivan	3.0†
Los Angeles Aqueduct	1909–11	27,052	Granite	Leyner	1.6
Moffat	1925	16,000	Gneiss and Schist	Ing.-Rand	2.5
Strawberry	1909–12	19,000	Sandstone and shale	Sullivan	1.5
Rawley	1912	6,235	Andesite	Leyner	1.9

* Cost of machinists' labor for drill repairs is roughly one-quarter of the cost of repair parts.
† This item includes labor, which is not included in the other examples.
NOTE.—The Leyners were hammer drills; all the others, piston drills.

Drill repair costs may be more logically expressed in terms of cents per ton or cubic yard of rock or ore broken.

At the Moffat tunnel (Table IX), from its start until March 1, 1927, near its completion, 531,962 cu. yd. of rock were excavated, at an average drill-repair cost per cubic yard of 4.5 cents for labor and 18.5 cents for parts (about 2.2 cents and 9 cents respectively per ton).

In the mines of the Commerce Mining and Royalty Co., Oklahoma (communication to author), the average total drill-repair cost in 1928–1929 was about 1.6 cents per ton broken. This covers the work of about 100 Gilman hammer drifter drills, used for rather severe service in drilling 20-ft. stope holes, together with a few light, one-man drills.

Table X gives the average cost of repair parts for five types of hammer drill used in the New Jersey Zinc Co.'s mine, Franklin, N. J. The figures, covering some years' operation, to Jan. 1, 1930, are for new parts only, not including labor in making repairs. The item "machine-months" is the total time elapsed since purchase of the drills, regardless of the time they were actually in operation.

TABLE X.—COST OF REPAIR PARTS FOR HAMMER-DRILLS, FRANKLIN MINE
(C. M. Haight)

Kind of drill	Number of drills	Machine-months	Cost per drill per month
Small drifter	28	1,851	$5.85
Heavy drifter*	18	1,798	3.58
Stoper	12	616	8.72
Blockholer†	77	4,951	5.78
Blockholer†	21	564	1.80

* These drills were less constantly used than the others, which accounts in part for their lower repair cost.
† A number of new pistons were not charged for by the makers; otherwise the cost would be higher.

CHAPTER XXI

DRILL-BITS AND MECHANICAL SHARPENERS

The drill steel, with its cutting edge, though not an integral part of a rock-drill, must in some of its features be considered in connection with the drill itself. This is especially true of the hollow steel so largely used for hammer drills, since the jet of air or water for removing the sludge or cuttings is transmitted from the axial tube of the machine directly through the bit to the bottom of the drill hole. This chapter will therefore deal with bits and sharpeners: with the mechanical details of bits, with their shanks as shaped to fit the drillchuck and to receive the jet of cleaning fluid, and with mechanical sharpeners, which are themselves operated by compressed air.

Other matters of prime importance concerning drill-bits are those relating to the quality and temper of the steel, and to the details of its heat treatment in forging and tempering bits to suit the varying requirements of their service. These subjects have chiefly to do with the metallurgical characteristics of the steel, as affecting its suitability for drilling in different rocks or ores and under different local conditions. They are more closely associated with the details of drilling than with the construction and operation of compressed-air drills themselves, and are therefore not strictly within the scope of this book. For information respecting the quality, temper and heat treatment of drill steel, see: *Trans.* Am. Inst. Min. Engs., Vol. 58, p. 3, Vol. 66, pp. 733, 758, 779, 789 and 806, Vol. 69, p. 666, Vol. 70, p. 290, and Vol. 74, p. 666; also, *Eng. and Min. Jour.*, May 12, 1917, p. 823, May 19, 1917, p. 875, July 28, 1918, p. 156, July 8, 1922, p. 68, Apl. 14, 1923, p. 670, and June 2, 1923, p. 977.

Drill-bits.—The word "bit" is employed rather loosely by both drill-runners and blacksmiths. Though it may sometimes denote merely the cutting end, it is often used also to designate the entire length of the steel, including its shank and cutting edges. The author prefers the latter designation, believing it to be accurate as well as convenient. By many drill-runners the "bit" is also called a "steel."

In Table I are given the dimensional details of the usual forms of drill steel, comprising the following cross-sections: solid octagon, solid

and hollow hexagon, hollow round, solid and hollow quarter-octagon and cruciform. The size of hexagon and octagon steel is measured by the distance between parallel sides of the section. Cruciform steel, though not often used, has the advantage of reducing the amount of labor in forging and shaping the cutting edges of the bit. In quarter-octagon steel, the length of the short side of the section is one-quarter of that of the long side.

Fig. 1 shows the different forms of bit in general use for rock-drills. Many other shapes and combinations of cutting edges have been devised, some of them quite complicated, with the idea of increasing the speed of drilling or removing the cuttings more effectually. But the simpler

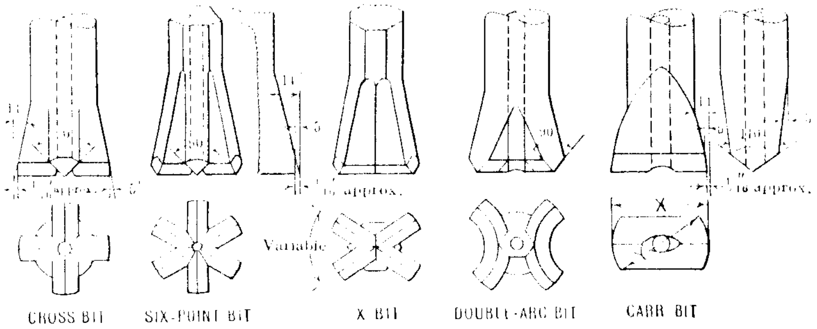


FIG. 1.—Types of Drill-bits in General Use.

forms are generally preferable; they do equally good or better work and are more readily sharpened. It should be noted, however, that, when mechanical sharpeners are employed, the last-named consideration is of less importance than when the bits are forged and sharpened by hand.

For *piston drills*, the standard forms of cutting edges are the right-angled cross and the X. The Z bit has been occasionally employed with some success, but, unless very carefully tempered, the ends of the cutting edges are apt to chip or to dull prematurely. It is best used with low air pressure and in soft rock. In solid, homogeneous rock or ore, the simple chisel-shaped bit, similar to that for hand drilling, is often found satisfactory.

For *hammer drills*, the six-point or "rose" bit is commonly used for the smaller machines in shaft-sinking and other down-hole work. They are formed and sharpened only by mechanical sharpeners. But, in general, six-point bits appear to have no advantage over the simpler cross and X bits (which can be sharpened either by hand or mechanical sharpeners), especially when the drill is provided with automatic rota-

TABLE I.—SPECIFICATIONS OF DRILL STEEL

Solid octagon, size, in.*	Weight, lb. per ft.	Diagonal diameter, in.	Area of section, sq. in.
3/4	1.58	0.812	0.466
7/8	2.15	0.947	0.635
1	2.82	1.082	0.828
1 1/8	3.56	1.218	1.047
1 1/4	4.40	1.353	1.294
1 3/8	5.29	1.488	1.565
1 1/2	6.34	1.623	1.864
Hexagon			
7/8 hollow	2.08	1.010	0.601
1 hollow	2.77	1.154	0.804
1 1/8 hollow	3.54	1.298	1.018
1 1/4 hollow	4.34	1.443	1.241
7/8 solid	2.25	1.01	0.663
1 solid	2.94	1.154	0.866
1 1/8 solid	3.73	1.298	1.095
1 1/4 solid	4.60	1.443	1.351
Cruciform		Short diameter, in.	Thickness of rib, in.
1	2.375	3/4	7/16
1	2.5	25/32	1/2
1 1/8	2.75	25/32	7/16
1 1/8	3.125	29/32	9/16
1 1/4	3.375	13/16	7/16
1 1/4	3.5	29/32	9/16

Quarter-octagon

	Size and section, in.	Weight, lb. per ft.	Length of short side, in.	Diagonal between short sides, in.	Diameter of hole in steel, in.	Sectional area, sq. in.
Solid	3/4	1.80	5/32	29/32		0.538
	7/8	2.45	3/16	1 1/16		0.732
	1	3.20	7/32	1 13/32		0.952
	1 1/8	4.06	15/64	1 23/64		1.211
	1 1/4	5.09	17/64	1 1/2		1.492
	1 3/8	6.05	9/32	1 21/32		1.811
	1 1/2	7.22	5/16	1 13/16		2.152
Hollow	3/4	1.60	5/32	29/32	5/16–9/32	0.5105
	7/8	2.19	3/16	1 1/16	9/32–5/16	0.6704
	1	2.98	7/32	1 13/64	9/32–5/16	0.8900
	1 1/8	3.80	15/64	1 23/64	5/16–3/8	1.134
	1 1/4	4.74	19/64	1 1/2	3/8–7/16	1.382

Hollow Round Steel

Size, in.	Weight per ft.	Diameter hole, in.	Net area, sq. in.
1	2.50	9/32–5/16	0.7233
1 1/8	3.21	5/16–3/8	0.9173
1 1/4	3.90	3/8–7/16	1.1166

* Larger sizes of solid octagon steel are also obtainable.

tion. For such drills, the chisel-edge Carr bit, considering its good cutting qualities, small loss of gage due to its large outer reaming and wearing surface, and the ease with which it is forged and sharpened, is excellent for hard rock and ore. It has been found that this bit is somewhat more liable than the cross bit to become wedged in the hole, when drilling in soft or fitchery ground. For solid, homogeneous rock, however, this bit is "free drilling." The double-arc bit also has its advocates.

On the whole, the cross bit, with a 14° to 5° outer taper (see Fig. 1), seems to be superior to all others for high-air pressure and hard rock. In soft rock, other characteristics, relating to the shape of the cutting edges, may make one bit better than another as regards its "mudding" qualities (keeping the bottom of the hole clear of sludge) and its freedom from tendency to "fitcher" or stick in the drill hole.

The wings of the cutting edges of the cross, X and rose bits, should be thick enough (usually 0.4 to 0.65 inch) to stand up well, but as thin as is consistent with strength, in order to leave space for free ejection of the sludge from the bottom of the drill hole. The angle forming the cutting edge of each wing usually ranges between 90° and 110°, an angle of 100° representing good practice under normal conditions. If the angle is less than 90°, the bit may cut faster, but it dulls sooner and is more apt to chip or break; if much greater than about 105°, the bit tends to crush the rock, when of average hardness, rather than fracture it. As a general rule, the smaller angle is best for the softer rocks; the larger, for hard rock. Variations in the proportions of the wings and design of the cutting edges are shown in Fig. 1. To facilitate regular rotation of the bit, and minimize rifling of the drill hole, the outer ends of the wings (reaming edges) should coincide with a circle described from the bit's center. To do good work, the cutting edges must be symmetrical (most readily secured with mechanical sharpeners); this equalizes wear and also tends to prevent rifling of the hole and consequent "fitchering" of the bit, which causes abnormal strains, with danger of breaking the bit or machine.

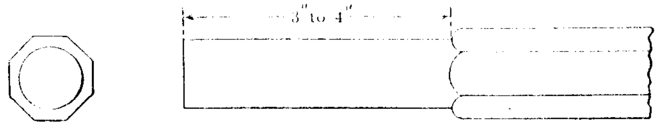
There should not be too great a difference between the diameter of the steel and the gage of the bit's cutting edges; a large difference weakens the tips of the wings. The ratio of steel diameter to bit gage normally varies from say 1 : 1.5 to 1 : 1.7. To provide ample support for the wings, their outer angle, or taper, from the steel bar to the forged-out wing tip of a new bit, should not exceed about 15°. With a double taper, beginning at 14° or 15°, and reduced to say 5° on approaching the wing tip, the bit stands up better than with a straight taper. Both designs are shown in Fig. 1. As a bit is shortened by wear and repeated resharpening, the angle of the taper for a given gage obviously increases, causing greater liability to fracture of the wings.

In drilling a hole deeper than say 15 or 18 inches, two or more steels, comprising a "set," are required. By the wearing away of the wing tips, the gage of each bit is reduced in drilling its share of the hole. Therefore, the first bit, or "starter," has the largest gage of a "set," that of each of the successive "follower" bits being smaller. In sharpening a set of bits, the blacksmith uses a series of ring gages, the relative diameters of which will depend upon the amount of wear taking place in drilling a given rock. The gage interval, or change of gage of the successive bits, is normally $\frac{1}{16}$ to $\frac{3}{32}$ in. A competent blacksmith carefully watches the condition of the dulled bits returned to him for resharpening, and varies the temper, and sometimes the gages of the "follower" bits of a set, as he judges necessary. He gets best results after having had experience in dealing with the local conditions as they may be presented. Obviously, with hand-sharpening on an anvil, there is a much greater degree of individuality in carrying out the details of this work (aside from questions of tempering) than when using machine sharpeners.

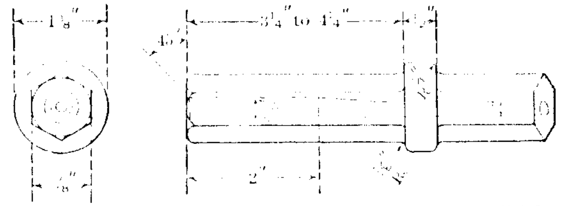
For a large mine or surface rock excavation, it is advisable to maintain some system of records of the performance of both drills and drill bits. These records are based upon reports from each machine, showing: footage drilled; number of steels received, resharpened and sent out from the blacksmith shop; breakage of steel and bits; number of bits cut off; reclamation, or welding of pieces of steel that might otherwise be discarded; and welding of fractured steel. Microscopic examination of broken steel often leads to useful conclusions as to the most suitable quality. Such data show what each machine is actually doing; the work of the blacksmith and drill-runner is thus more closely correlated, greater economy is induced in the consumption of steel, with improved general efficiency. This matter was referred to in Chapter XX.

Breakage of Drill-steel became a more serious problem after the introduction of hammer drills, due to differences in working conditions of the bits. No clear-cut reasons can be assigned for this increased breakage. No doubt it was partly due to the fact that, for many of the smaller sizes of hammer drills, lighter steel and smaller bit gages are desirable than those used with piston drills. But, in recent years, manufacturers have succeeded in making drill-steel alloys that give better service. Much has been published on this subject in the *Trans.* Am. Inst. Min. Engs. and the technical journals. For details of a useful investigation, see *Trans.* A. I. M. E., Vol. 66, p. 758. As the heat treatment of the steel, in forming bits and shanks, is intimately related to questions of breakage, blacksmith shop practice should receive careful consideration.

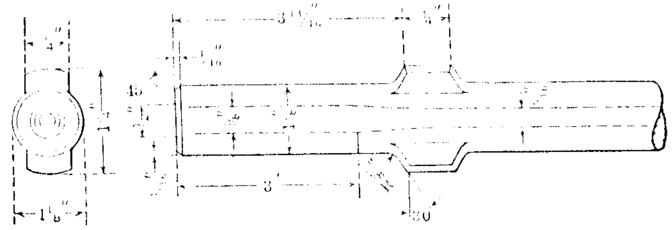
Shanks for Drill-bits are shown in Fig. 2 in their usual forms for both piston and hammer drills. The actual dimensions of shanks depend upon the size of the drill and of the steel used. In piston drills, the shanks are clamped firmly in the chuck by bolts; in hammer drills, the shanks of both solid and hollow bits are not clamped, but held loosely in the chuck. The clearance between chuck and shank is normally $\frac{1}{64}$ in. By reference to the longitudinal sections of hammer drills (Chap. XIX) it will be seen that the end of the water tube enters a short distance into the hole in the hollow steel. To avoid injury to the water tube, the shank must be carefully forged, so as to center the hole accu-



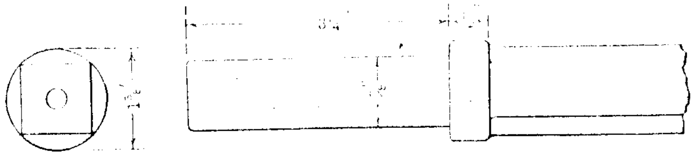
ROUND SHANK FOR PISTON DRILLS



HEXAGON SHANK WITH COLLAR, HAMMER DRILLS



LEYNER LUG-SHANK, HAMMER DRILLS



SQUARE SHANK, HAMMER DRILLS

FIG. 2.—Drill-bit Shanks.

rately and prevent the shank from coming into actual contact with the tube.

Table II gives the lengths of the original bar of drill steel required for forming the shank and bit. These data serve for estimating the amount of bar stock to be ordered for a given job.

TABLE II.—LENGTHS OF BAR STEEL REQUIRED FOR FORMING SHANK AND BIT

Section: hollow and solid hexagon, round, octagon or cruciform, diameter, in.	Length of Stock for Forging Shank, Inches: 3¼-in. Jackhammer shank	1¼ in. Jackhammer shank	Leyner Ingersoll lug shank	Stoper drill shank	Piston drill steel: With shank	Piston drill steel: Without shank
⅞	7	8		6	4¾	5
1	6¾	7¼	8	6	5¼	5½
1⅛		7⅜	7⅝	6	5¼	6
1¼			7½	6	5¼	6

Length of Stock for Forging Bit, In.

Section, diameter, in.	1½	1⅝	1¾	1⅞	2	2⅛	2¼	2⅜	2½	2⅝	2¾	2⅞	3
Cross and Carr bits													
⅞	1½	2	2½	3¼	4¼	5¼	6¼	8¼	10¼	12¼	14¼	16	17
1	⅞	1⅜	1¾	1¼	2	2⅜	3¼	4¼	4¼	6	7¼	8¼	9
1⅛	¾	1	1¼	1½	1¾	2	2½	3⅜	3⅝	4⅜	5½	7½	8¼
1¼		½	⅝	¾	1	1¼	1½	1¾	2¼	3¼	4	5	5¾
Six point rose bit													
⅞	1⅞	2⅝	3¼	4¼	5¼	6⅛	8¼	9½	12	15¾	17¼		
1	1⅝	1½	1¾	2¼	3	3¼	4¼	5¼	7	8⅞	9⅜	10¼	11½
1⅛	⅞	1⅜	1½	1⅞	2⅜	2⅞	3¼	3⅞	4¼	6½	7½	8¼	9

To find length of bar required for any length of "steel," add to the lengths given in table for shank and bit, the "drilling length" for the given depth of hole.

Lugs or collars (see Fig. 2) are in general use for hammer-drill bits; they retain the bit in the chuck and aid in pulling fitchered bits from the drill hole. For machines drilling holes directed steeply upward in free-drilling ground, bits sometimes have neither lugs nor collars. In such ground the Carr bit will work satisfactorily without lugs.

Aside from fractures of the bit wings, breakage of drill steel seems to be most liable to occur at the point where the shank is forged down from the original steel bar; probably due to the effect of the forging heat upon the temper of the steel. In shanking hollow steel, whether by hand or a mechanical sharpener, a closely fitting pin is inserted in the hole in the steel to maintain its shape while the shank is being forged; or a pin is driven to open a partly closed hole. The orifice in the shank for accommodating the end of the drill tube should be $\frac{5}{16}$ to $\frac{3}{8}$ in. in diameter, and opened at this diameter to a depth of at least 3 in. (see the dotted lines in Fig. 2).*

Mechanical Drill-sharpeners.—Drill-sharpeners operated by compressed air, have become in recent years increasingly important in connection with rock-drill work. Where only a few drills are operated, hand sharpening is still properly practiced; but at large mines, and in general for large-scale rock excavation, mechanical sharpeners are unquestionably effective and economical. The earliest mechanical drill-sharpener appeared about 1907. They are now made by all the leading rock-drill manufacturers. Their use conduces not only to rapidity of work in the blacksmith shop, but also to greater accuracy and symmetry in the shaping of bits and shanks, and more uniformity in the subsequent heat treatment.

The operations performed by drill-sharpeners may be outlined in general terms as follows: After heating to forging temperature the end of the length of bar steel on which the bit is to be formed, the steel is firmly clamped in the sharpener by a pair of heavy jaws (a vise), the movable member of which is actuated by the piston of a powerful compressed-air cylinder. Thus held in position, the heated end of the steel is "upset" by blows of a horizontal air-hammer, constituting a part of the sharpener and usually resembling a rock-drill. The cutting edges of the bit are opened by means of a "dolly" (mold), placed against the hot upset end of the steel, and struck a rapid succession of strong blows by the hammer. In most drill-sharpeners, the partly formed bit is then swaged and dressed to the desired gage by a second air-hammer; after which it receives its appropriate temper. In making a new bit, or resharpening a dull one, the steel is usually shifted from one hammer to the other, being thus alternately upset and swaged, until the wings are of accurate shape and gage. A competent smith can finish a bit in one heat.

Shanks are formed by the sharpening machine as a separate operation. For this, accessory parts are provided, for clamping the steel in position while the shank end is being upset and shaped. Shanks can be either upset or swaged with a single heat; for both operations, two heats are necessary. It may be added that other accessory parts are furnished by the makers, for forging a variety of devices and tools used in connection with rock excavation. It is worthy of note that the hammers performing the major operations of mechanical sharpeners are modified rock-drills, generally of the hammer-drill type.

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* For further details regarding shanks, see *Trans.* Am. Inst. Min. Engs., Vol. 66, p. 730.

Further details of the operation of mechanical sharpeners are given in connection with the following examples, which have been chosen to illustrate the essential features of different makes.

Leyner sharpeners are made by the **Ingersoll-Rand Co.** in several styles and sizes, models 34, 4 and 50 now (1929) being the standard designs. Figs. 3 and 4 show the large No. 50 model, weighing 2935 lb.; height, 54 in.; floor space, 48 by $38\frac{3}{8}$ in. This machine makes bits of standard forms to a maximum gage of $3\frac{3}{8}$ in. on hexagon, octagon or quarter-octagon steel up to $1\frac{7}{8}$-in. diameter, and also on cruciform or spiral auger steel to 2-in. diameter.

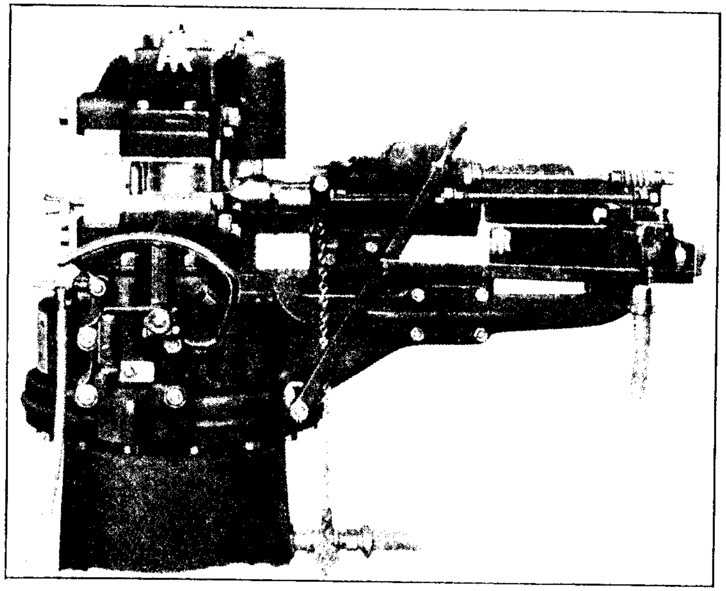


Fig. 3.—Leyner Drill-sharpener, Model 50 (Throttle-lever side).

Referring to Fig. 4, the dies *a*, for holding the steel on which the bit is to be formed, are set in the renewable die seats *b*, which in turn are fitted to machined seats in the upper and lower jaws *c* of the compressed air vise. The upper, movable member of the vise consists of the crosshead *d*, piston- or crosshead-rod *e*, and the differential piston, *f*, *g*. This piston, which actuates the crosshead, is clamped against a shoulder on the piston rod by lock-nut *h*. The stationary part of the vise comprises the base *i*, and the cylinder *j* of the differential piston.

Compressed air for operating the sharpener, entering at *o* in the base, acts constantly upon the small diameter *f* of the differential piston, which tends to keep the crosshead in its upper position. The upper side of the large diameter *g* of the piston is then open to atmosphere for exhaust. But, by moving the throttle-lever *p* one-sixteenth of a revolution, valve *q* is raised, thus closing the exhaust and admitting live air to the upper side of *g*. This forces the piston and crosshead down, against the constant pressure acting on the small diameter *f* of the piston.

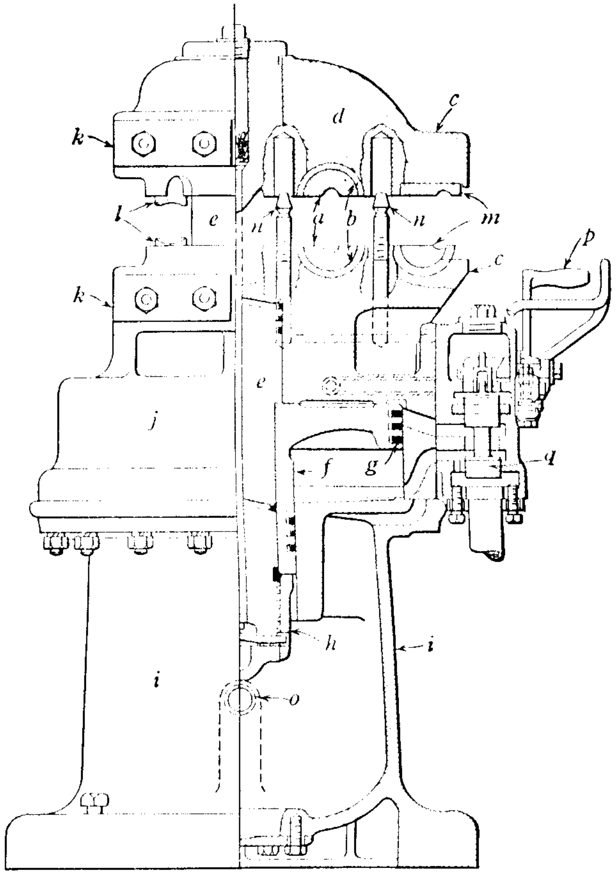


Fig. 4.—Leyner Drill-sharpener, Model 50 (vertical section at right angles to view in Fig. 3).

On the right-hand side of Fig. 3 is the horizontal compressed-air cylinder, in the end of the piston rod of which is socketed the dolly for forming the cutting edges of the bit. Dollies are furnished for any desired shape of bit (see Fig. 1); that for the ordinary cross bit is shown by Fig. 5, in its relation to the steel lying in its die. Scale forming on the hot bit while being shaped is removed by the two "scale-blowers" (*n*, *n*, Fig. 4). A blow-gun is also provided for blowing out hollow steel.

Referring again to Fig. 4, the main cylinder casting *j* carries the lower halves of dies *a* and of all parts of the accessory or auxiliary equipment; these parts being held in place by clamp-plates *k*. The accessories include several devices, such as the shearing knife *l*, for cutting and trimming the hot steel, and the automatic centering device *m*. The latter holds and alines a bit or shank for punching. The lower member of *m* is a semi-cylindrical block, which is rotated on its axis by the descending crosshead until the side of the block opposite the groove in which the steel lies comes into contact with the crosshead, thus gripping the steel. Steels of different sizes and sections are in this way clamped, centered, and alined with the punching pin. The punching operation is for opening up the holes in bits or shanks of hollow steel, which have been deformed or partly closed during forging.

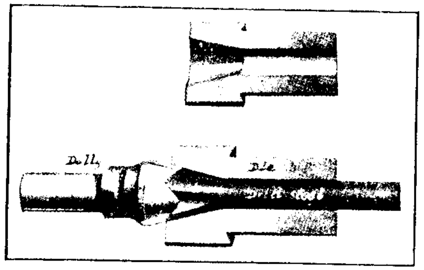


Fig. 5.—Dolly in Position for Shaping a Bit.

All moving parts of the sharpener are lubricated by a "heartbeat" lubricator (see Chapter XIX, Fig. 66 and description). This, with its filter, is in the upper end of the throttle-valve chest *q*, the oil being circulated by pressure of the compressed air.

The smaller models 4 and 34 sharpeners, weighing 1,750 and 1,430 lb. are not provided with shearing knives nor centering device. They make bits to a maximum gage of 2½ in., on 1⅛-in. steel.

Sullivan Machinery Co. makes two types of drill sharpener, differing substantially in design from the preceding. Model A, weighing 3,800 lb. net, is intended for bits ranging from 1⅛ to 3½-in. gage, on all standard sizes of steel. The small Class C machine, weighing 1,100 lb., makes bits from 1⅛ to 2½-in gage, on ⅞ or 1-in. steel.

Figs. 6 and 7 show the heavy, Class A sharpener, 73 in. high, 61 in. long and 29 in. wide. Referring to Fig. 7, the frame or pedestal contains the horizontal cylinder *a*, 12 in. in diameter, which operates the air vise for clamping the steel while being upset and worked. On the crosshead block *b* is loosely hinged the toggle *c*, which, on the forward stroke, thrusts downward the bottom yoke *d*. This yoke is connected with an upper yoke, forming the movable member of the vise, by a pair of heavy vertical siderods sliding in long guides bored in the frame (one of these rods is shown in Fig. 6). The action of the vise is controlled by a hand-operated valve, in a chest on the side of the frame.

There are two hammers, both modified Sullivan hammer drills. The horizontal hammer *e*, which has a floating piston, upsets the steel and forms the bit's cutting edges. For the latter operation, the piston strikes a pin projecting from the front cylinder head, the end of the pin being shaped to carry the dolly shank. The vertical hammer *f* resembles a piston drill. This swages and dresses the bit, after the cutting edges have been opened by hammer *e*, striking a light or heavy single blow, or successive blows, as required. Air is admitted to both the clamping cylinder and the upsetting hammer by movements of the same hand lever. A separate branch of the inlet pipe carries air to the swaging hammer. The machine has the usual blow-gun for cleaning out hollow steel, and air jets for removing scale.

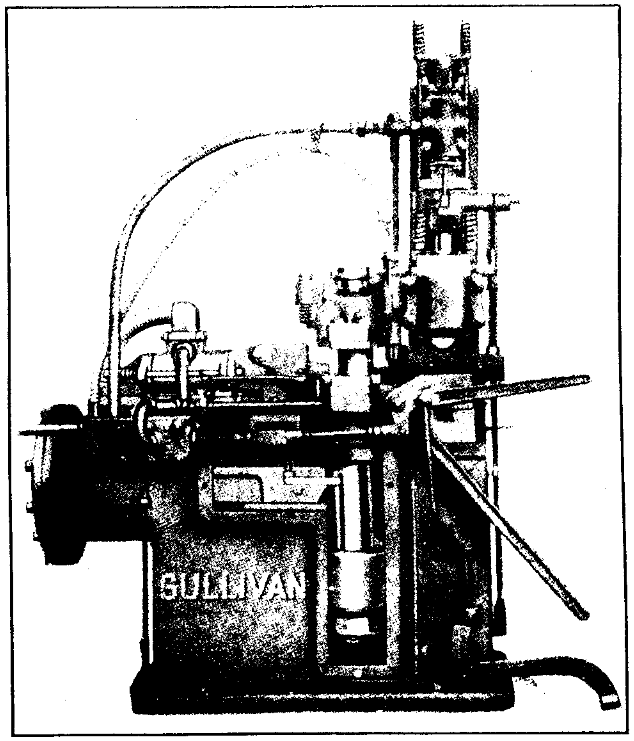


Fig. 6.—Sullivan Class A Drill-sharpener (side cover removed).

To open the hole in shank or bit on hollow steel, this sharpener is provided with an accessory puncher. This may be either a small air-hammer, mounted on the side of the frame, and which operates without the use of the vise, or a special power retracting-punch and pin. The latter drives the pin while the steel is held in the vise, the punch cylinder being moved up to the clamped steel by a hand lever. This device is so designed that, after the hole has been opened to a sufficient depth, further movement of the lever opens an air valve behind the retracting piston, thus forcing it against the cylinder end and knocking out the punch pin by a single blow. In this description some of the mechanical details are omitted.

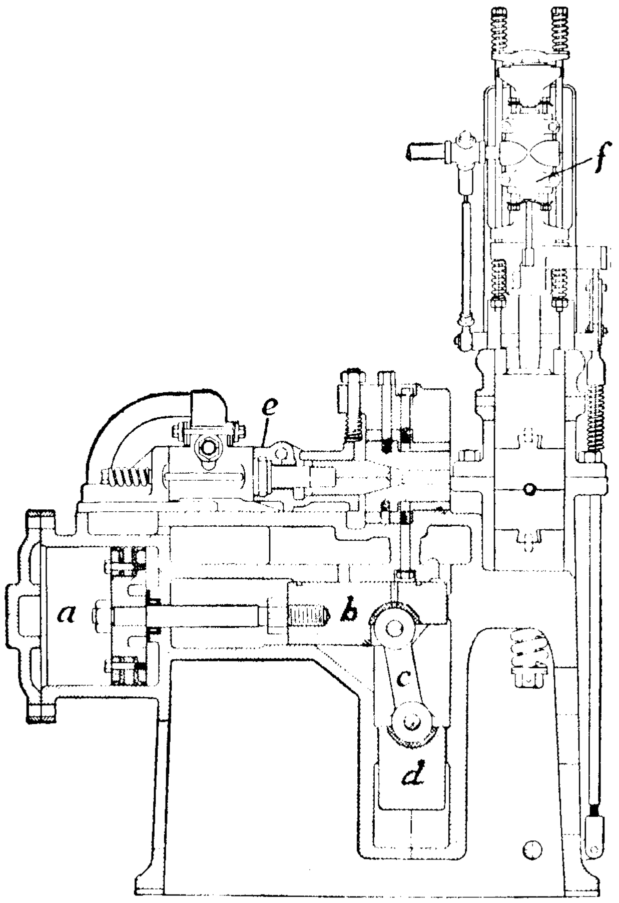


Fig. 7. Sullivan Class A Drill-sharpener (vertical section).

Class C sharpener is a much smaller, portable machine, that may be mounted on skids (floor space, 30 by 42 in.), but has both horizontal and vertical hammers. Its clamping lever, pivoted near the top of the frame, is opened and closed by a toggle operated by a small air cylinder. In other respects, this sharpener is similar to the Class A.

Holman Bros.' drill-sharpener, some of the details of which are quite different from those in the foregoing examples, is shown diagrammatically by Fig. 8. It has a horizontal, semi-valveless dollying hammer (not shown in the illustration), at the back of the machine, in line with the gripping dies of the vise at *G*, and a vertical swaging hammer *H*. The latter, of the automatic-valve type, is operated by a foot lever at the base of the pedestal.

By means of the air piston *P*, and the lever *L* working on the fixed center *C*, the block *B* is caused to slide horizontally to the right or left. The block is held accurately in alinement by a pair of heavy guide rods, one of which is at *D*, and it carries the movable parts of the dies *G* and *F*. Dies *G* hold the steel while the bit is being upset and roughly shaped. In line with the gripping dies *F* is set a valveless punching hammer, for opening the holes in bits or shanks of hollow steel. The movements of the dollying hammer and of the sliding block *B* are controlled by a single valve *V*, operated by the hand lever *T*. Either of the dies *G* and *F* may therefore be used, as required, but not both at one time.

In view of the preceding descriptions, further details of this sharpener are unnecessary.

Mechanical drill-sharpeners are also made by the **Gardner-Denver Co., Gilman Manufacturing Co., Hardsocg Wonder Drill Co.,** and the **Climax Rock Drill and Engineering Works.**

Forging Furnaces for Drill-steel are made by most of the prominent rock-drill manufacturers. The simpler forms consist of a rectangular, cast-iron box, lined with firebrick, having a gas or oil burner at one end, and mounted on four wrought-iron pipe legs or a cast pedestal. Their net weight ranges from about 1,000 to 1,500 lb. The bits, say 10 to 15 in a batch, are held in position for heating by an adjustable support, attached to the furnace or standing on the floor.

Oil-fired furnaces are not only economical and convenient, but they can be regulated to heat the steel more accurately and uniformly than is possible in an ordinary, coal-fired forge. To indicate the temperature of the furnace, some form of pyrometer is desirable.

Gardner-Denver Co., for example, furnishes a small electrical pyrometer, the "comparascope," which can be adjusted for a heat of from 1000° to 1800° F., in 50° graduations.

Water is used for tempering bits; shanks are best tempered in a tank of oil, kept cool by surrounding it with circulating water.

The Ingersoll-Rand Co. makes the oil-fired furnace shown in Fig. 9. Its lining is of standard 9 by $4\frac{1}{2}$ by $2\frac{1}{2}$-in. firebrick and 9 by $2\frac{1}{4}$ by $2\frac{1}{2}$-in. soapstone brick, laid in thin mortar, with close joints. A layer of asbestos cement is placed between the bricks and the iron casing. The auto-

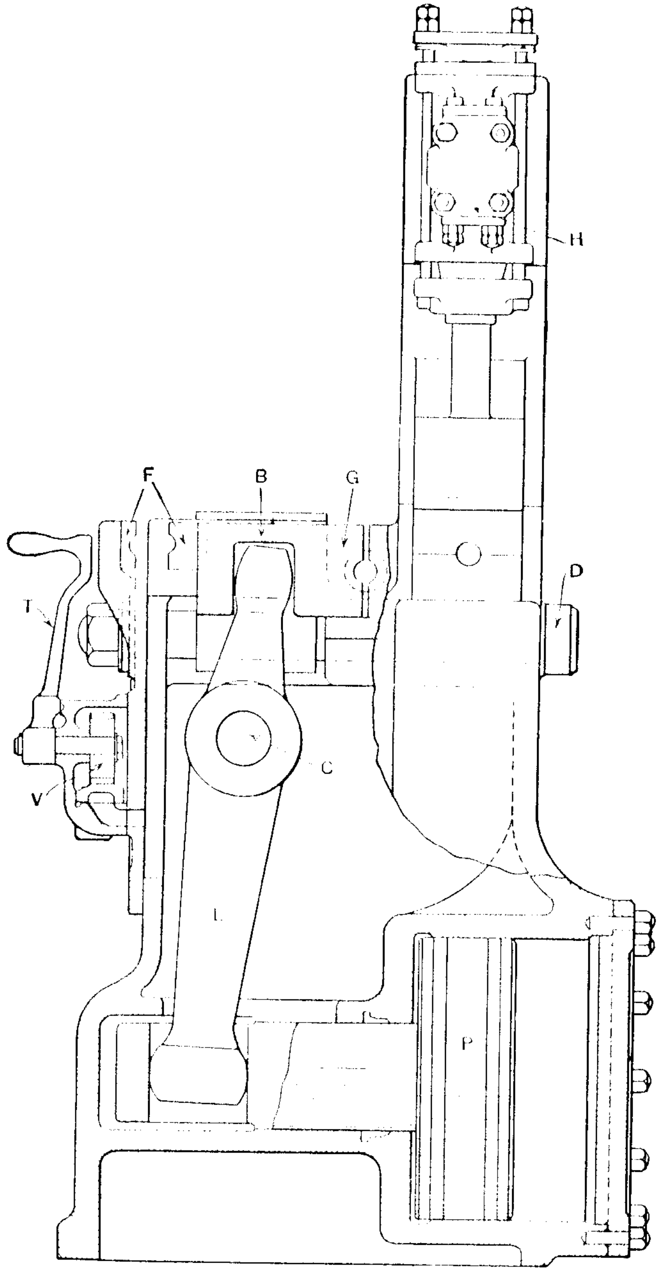


Fig. 8.—Holman Drill-sharpener (diagrammatic).

matic temperature control consists of a pyrometer and a thermocouple, the latter being set to extend through a hole in the lining about 1½ in. into the furnace. Care must be taken to connect the thermocouple positive and negative wires correctly with the corresponding binding posts of the pyrometer. The pyrometer, provided with colored 40-watt signaling lamps, is operated by a Dickson control motor, using alternating current. If direct current only is available, a rotary converter is necessary, usually working at 110 volts and 60 cycles.

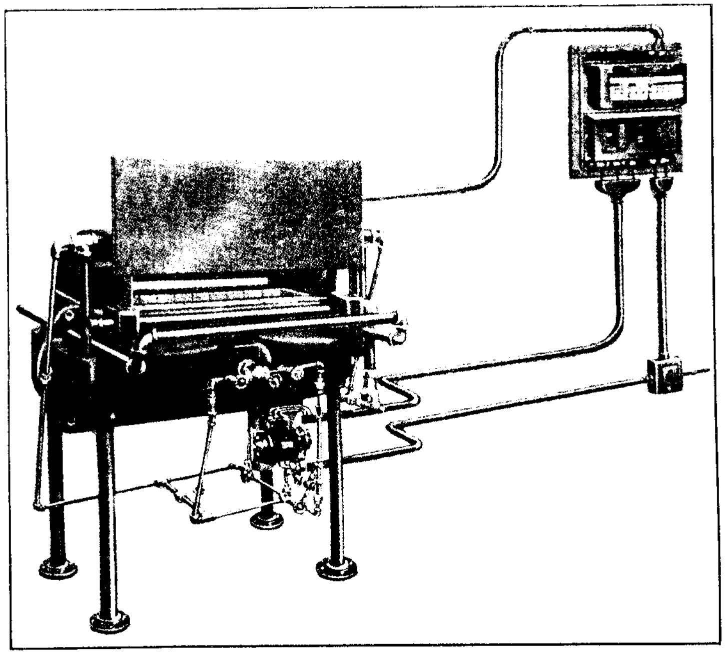


Fig. 9.—Ingersoll-Rand Co. Oil-fired Drill-steel Furnace.

Fig. 10 shows three typical heating curves. Standard Ingersoll-Rand drill-steel should not be forged below 1800°, nor heated above 1950° F. Curve 1 indicates too high a temperature, say 2000° to 2200° F.; consequently the steel must be removed from the furnace below point *A* on the curve, and, due to the excess of furnace temperature above the required temperature of the steel (see horizontal line), there may be trouble from cracking while forging and hardening. With curve 2, the furnace is being run close to the correct temperature, and the steel is removed at a point on the curve above *A*, with only a small temperature difference. Curve 3 exhibits the results of slower treatment. This is usually the best method, as the steel is not removed until it reaches the furnace temperature.

Heat-treating and Tempering Machines.—The present-day tendency in the treatment of drill steel is to minimize the effect of the variable personal factor of the blacksmith. By employing furnaces of the semi-muffle type, the temperature of which can be regulated within quite narrow limits, and handling the steel during tempering by mechanical devices, subject to only a limited control by the operator, greater uniformity in the product is secured.

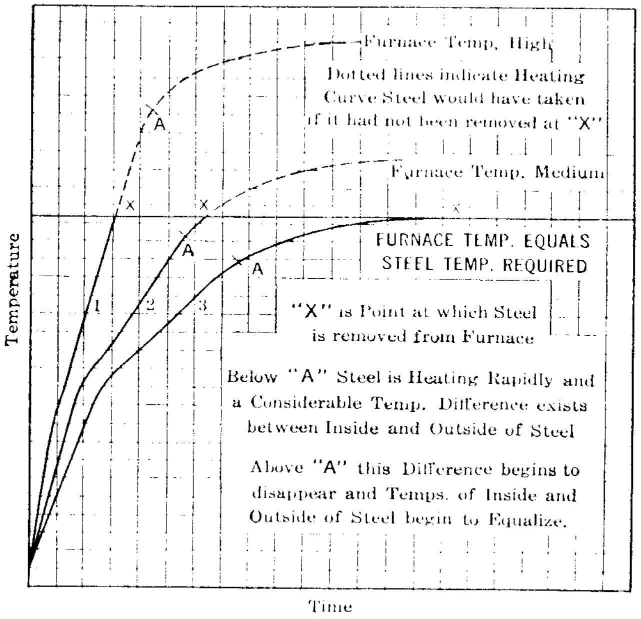


Fig. 10.—Typical Heating Curves of Furnace.

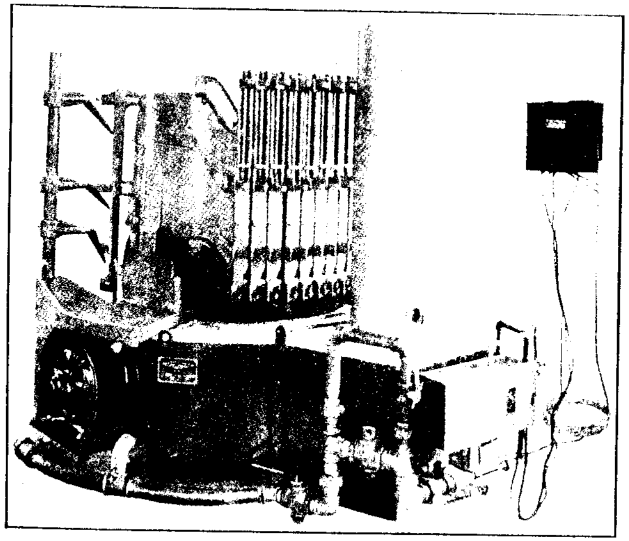


Fig. 11.—Gilman Heat-treating Machine for Drill-bits.

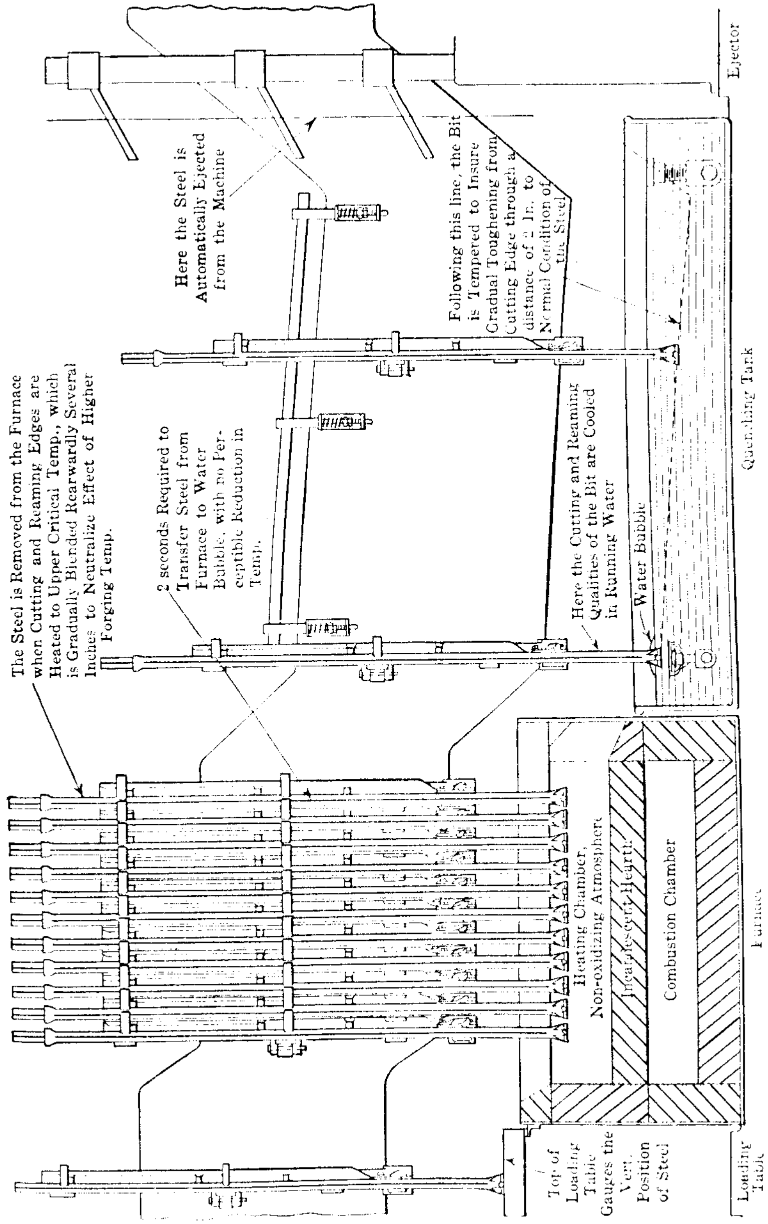

The Steel is Removed from the Furnace when Cutting and Reaming Edges are Heated to Upper Critical Temp., which is Gradually Blended Rearwardly Several Inches to Neutralize Effect of Higher Forging Temp.
2 seconds Required to Transfer Steel from Furnace to Water Bubble, with no Perceptible Reduction in Temp.
Here the Steel is Automatically Ejected from the Machine
Following this line, the Bit is Tempered to Insure Gradual Toughening from Cutting Edge through a distance of 2 In. to Normal Condition of the Steel
Here the Cutting and Reaming Qualities of the Bit are Cooled in Running Water
Water Bubble
Quenching Tank
Ejector
Heating Chamber, Non-oxidizing Atmosphere
Incandescent Hearth
Combustion Chamber
Furnace
Top of Loading Table Gauges the Vert. Position of Steel
Loading Table

Fig. 12.—Gilman Heat-treating Machine (diagrammatic).

Gilman Manufacturing Co. makes an elaborate machine shown in (Figs. 11 and 12), comprising a heating furnace and a quenching tank.

It is designed for use where more than say 300 bits per 24 hours are to be handled, and is operated by a 3-hp. electric motor. Most of its features are shown in the illustrations. The preliminary forging of the bits is best done by means of a separate furnace, as already noted. In connection with the machine, an automatic pyrometer control maintains the furnace temperature within a range of about 20° F.

Since the furnace resembles a muffle, the bits are heated in a reducing atmosphere, thus preventing decarburization of the steel. The temperature in the furnace is kept approximately 150° higher than the critical temperature of the steel. The bits are heated evenly, until they reach the upper critical range, and are then transferred to the quenching tank, for hardening and tempering. In the tank the rate of flow and level of the water are regulated by the operator, so that the precise depth of submersion of the bits, which is gradually increased from $\frac{1}{2}$ to $4\frac{1}{2}$ in., is under control. By these means, the successive steps in the treatment can be adjusted for the varying lengths of time required for steel of different kinds and bits of different gages.

CHAPTER XXII

COAL-CUTTING AND QUARRYING MACHINERY

COAL-CUTTING machines have almost entirely replaced hand labor in "under-cutting" the coal, preparatory to breaking it by blasting. Objects: (1) to mine cheaply; (2) to decrease the proportion of "fines" produced, thereby increasing the market value of the product; (3) to increase the rate of production from a given extent of mine workings. Coal cutters are used chiefly in bituminous collieries; in recent years they have also been successfully employed for mining anthracite in veins of rather flat pitch. They groove or undercut the face or breast of coal, generally close to the floor and to a depth of several feet. The mass so undercut is broken down by comparatively light blasts. Unless wages are very low, machine mining is always economical.

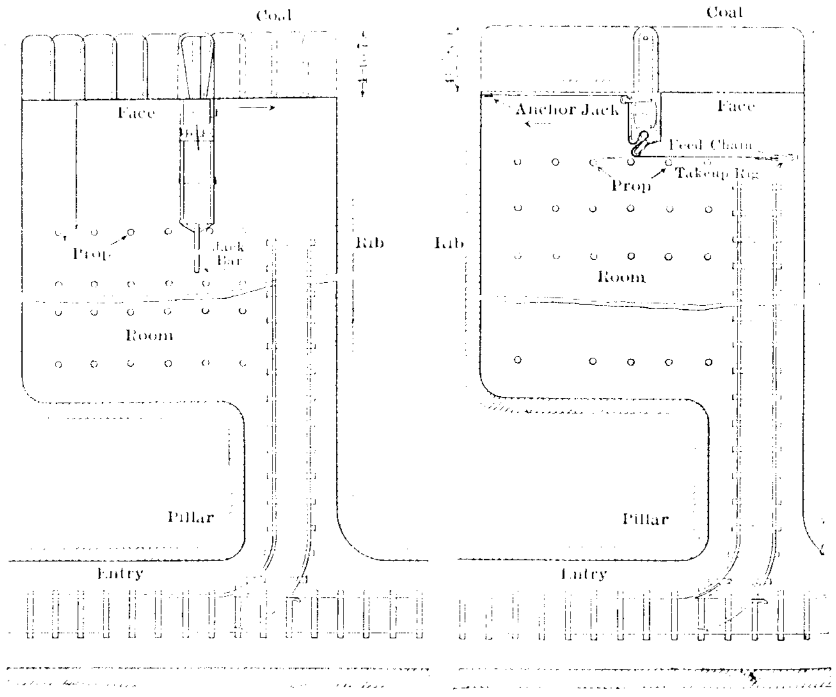
Coal cutters comprise four classes: (1) Endless-chain; (2) Rotary-bar; (3) Disk; (4) Pick machines. Machines of the first three classes are driven by either electricity or compressed air. Picks are operated by compressed air only, no entirely satisfactory electric-driven pick having yet been brought out. The bar and disk machines, obsolete in the United States, are still built by some European makers.

Endless-chain Cutters are mostly electric-driven, although compressed-air drive is furnished for use in gaseous mines, or where local conditions make it more convenient. Several of the American makes of chain machine are quite extensively used in Great Britain, France, Belgium and other European countries.

Chain cutters comprise "longwall machines," for longwall mining, and "shortwall machines," for room and pillar mining. The modes of manipulating them are shown in Figs. 1 and 2.

Longwall machines, and some types of shortwall also, are self-propelling and operate continuously along the face of coal to be undercut. They are fed forward by a chain or wire rope, winding respectively on chain sprockets or on a drum, mounted on the rear end of the machine, and driven by gearing from the motor engine. The chain or rope is made fast to a timber prop or an "anchor-jack," set between roof and floor, at any desired distance from the machine. By throwing the feed drum or sprockets into gear, the machine pulls itself along the face while making the undercut.

For rope-feed a single drum 16 or 18 in. in diameter is used, with friction clutch; for chain-feed, there is a group of several small sprockets, arranged for example as in Fig. 2. Each of these modes of feed has its advocates. Rope operates more smoothly, and is best on steeply inclined seams, but is subject to rapid wear on the small diameter drum. Chain-feed costs less, with relatively little wear, and is often preferred. Fig. 5 shows a double drum, for rope-feed. The part *k* is a "sumping" drum, by which the machine is brought into position at the point where cutting is to be started, and is then moved forward or "sumped" into the face a distance equal to the depth of the slice to be taken. The other part of the drum, *j*, is for feeding the machine along the face.



Room Mining with Old Style Breast Machine

Room Mining with Self-propelling Machine

Fig. 1.

The makers of coal cutters furnish low, special trucks, on small wheels, for moving the machines from place to place underground. Some trucks are self-propelled by a sprocket chain, driven from the cutter engine, others are hauled by mules.

For many years chain machines were driven by air engines of the reciprocating piston type, but they are heavy, with high cost of repairs, and have been superseded by the lighter turbine drive (Figs. 3, 4 and 5). This radical change has now extended to nearly all makes of the first three classes of coal cutters.

Jeffrey Manufacturing Co. makes the type 24-A longwall air-driven machine shown in Fig. 3, details being given in the succeeding illustrations. Referring to Fig. 5, the frame *a*, carrying the cutting chain, is pivoted on the front end of the machine. This chain runs on two sprockets, one at each end of the cutter frame. At the inner end is the driving sprocket, operated through worm gearing by the compressed-air turbine *h* (Fig. 4). By means of the jaw clutch *f*, the cutter chain can be started or stopped, as required.

The turbine consists of a pair of rotors, with helical blades or teeth cast on their surfaces, and designed to work with air pressures of 40–80 lb. The rotor teeth are accurately ground to each other, and make a close fit in the turbine casing.

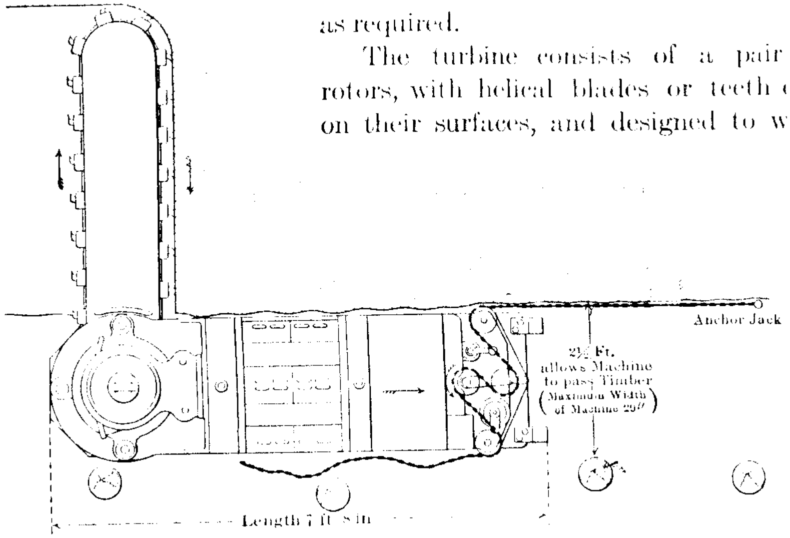


Fig. 2. - Position of Longwall Machine in Cutting along the Face.

A worm gear *i* (Fig. 5), on the opposite end of the turbine shaft *g*, operates the feed and sumping drums *j* and *k*, as follows: The feed worm-wheel *l* has a crank pin *m* and connecting rod *n*, which is pinned to the ratchet drive-lever *o*, carrying a pawl *p* engaging with the ratchet *q*. On the lower end of the vertical spindle *r* of this ratchet is a pinion *s*, engaging with a gear *t*, which is attached to the cone clutch *e*. The handwheel *u* operates the drums by forcing them down into contact with the cone clutch.

In the links of the cutting chain are sockets, in which the teeth or bits are held by set-screws. The bits are of different shapes, with their

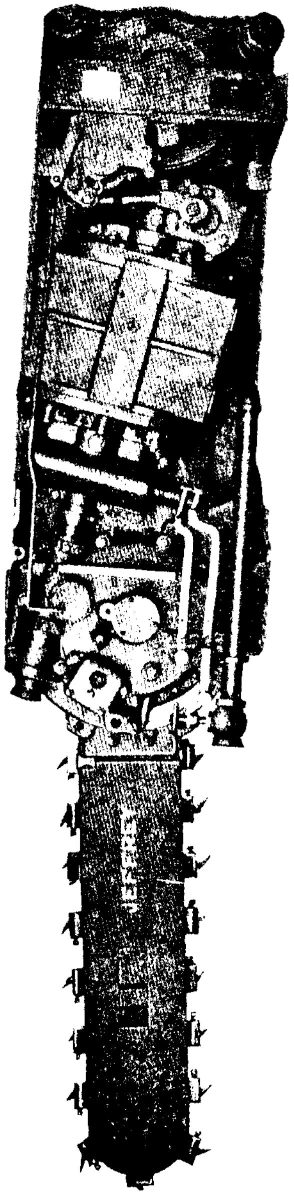


Fig. 3.—Longwall Air Turbine Coal Cutter, Type 24-A, Jeffrey Mfg. Co.

cutting edges at different angles, and are grouped as required for varying the thickness or height of the groove ("kerf") cut by the machine.

The Jeffrey longwall machine is 8 ft. 2 in. long, 31 in. wide and 19 in. high; weight, 6,000 lb.; lengths of cutter arm, 4, 5, and 6 ft. This company makes a "shortwall" cutter, type 35-B, for room work; also a new design of "arcwall" machine. Both are driven by air turbine. On the rear of the arcwall cutter is a large reel, carrying hose for conveying air to the machine when working at a distance beyond the end of the supply pipe line.

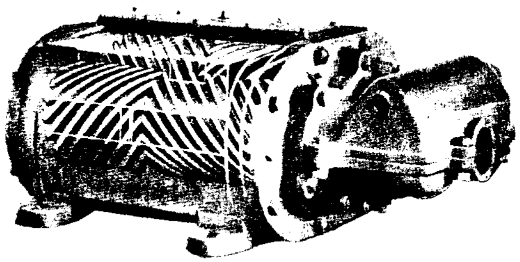


Fig. 4.—Compressed-air Turbine for Longwall Coal Cutter, Jeffrey Mfg. Co.

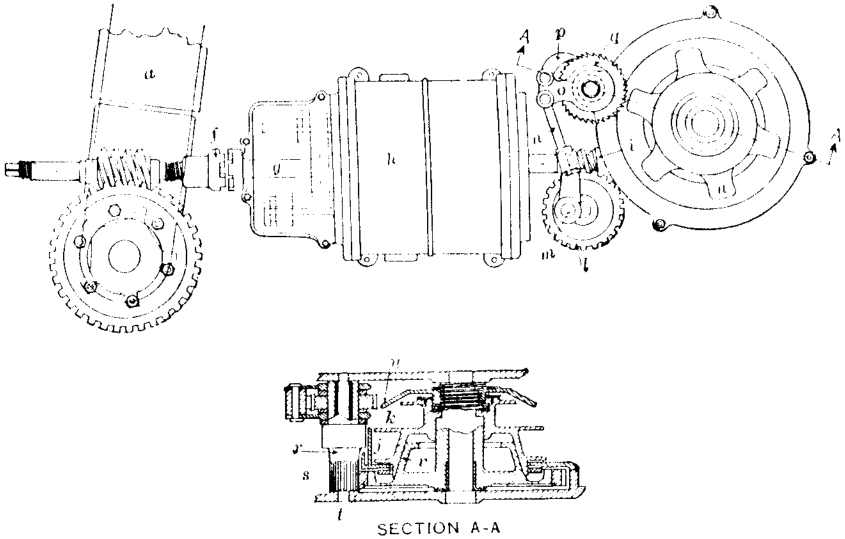


Fig. 5.—Diagram of Operating Mechanism, Jeffrey Coal Cutter, Type 24-A.

Sullivan Machinery Co.'s chain machines, usually driven by electricity, are also built with air-turbine drive. The turbine is the same in principle as that shown by Figs. 4 and 5, though differing in some of its details. Ball-bearings are provided for the rotors. For further description of the air turbine, see Chapter XVI, Fig. 5 and accompanying text. The current types of these machines are the CLA 2, for longwall mining, and the CE 7, for room-work (Figs. 6 and 7), specifications of which are given in Table I. Their mode of operation is shown by Figs. 1 and 2.

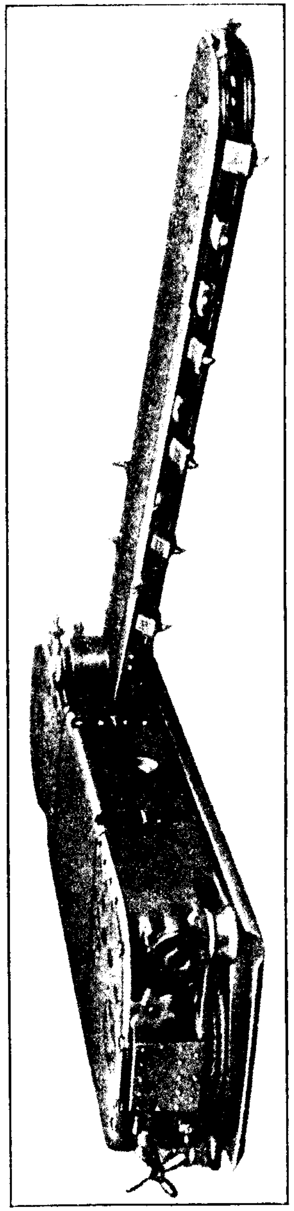


FIG. 6.—Sullivan Longwall Chain Machine, Class CLA-2, with 8-ft. Jib.

The CLA 2 machine is a recent design. For greater portability and convenience in making repairs, the body of the machine consists of three sections (respectively, feed, motor and drive), which are alined by dowels and bolted together. The machine's small height, 12 in., adapts it to work in thin veins; the feed, by rope or chain, is interchangeable; and the speed of the cutter chain can be varied to suit the conditions. The jib can be swung through an arc of about 240°. By inverting the drive section of the body, and mounting the machine on a steel frame skid, the groove can be cut at different heights above the floor of the seam.

Mavor & Coulson, Glasgow, Scotland, build a chain machine, driven by either electricity or compressed air. The compressed-air motor is of the turbine type, similar to that in Fig. 4, with differences in some of the mechanical details.

The frame or jib carrying the endless chain in which the cutting bits are set is pivoted to the forward end of the machine, and can be swung horizontally through an angle somewhat exceeding a maximum of 180° with respect to the body of the machine. As described in connection with Fig. 5, the main chain sprocket is driven by double-reduction spur gears and worm gearing.

For "undercutting," or making the groove (kerf) at the bottom of

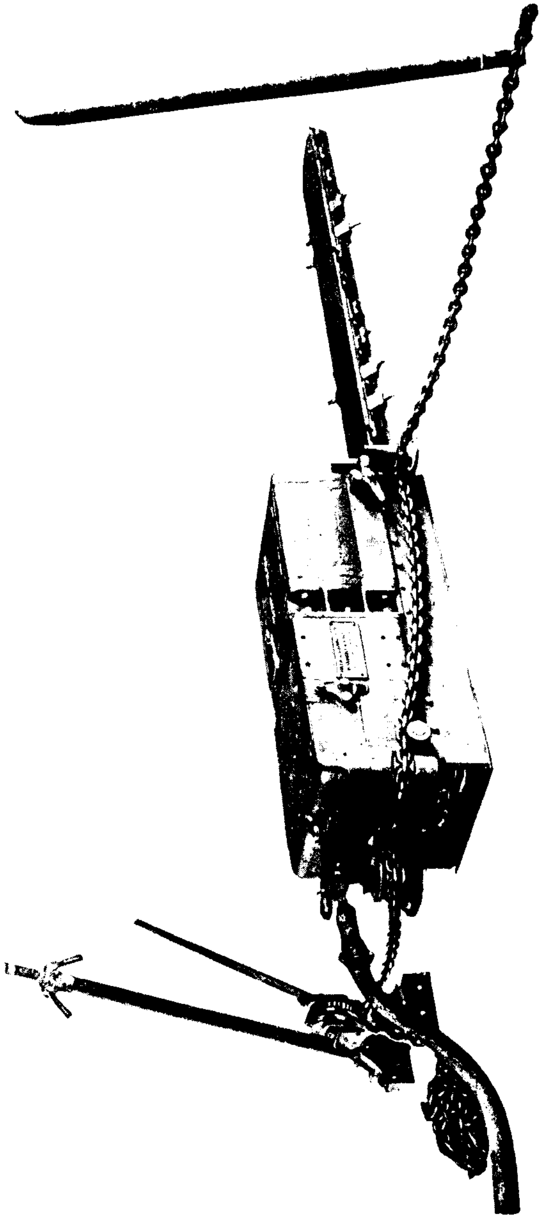


FIG. 7.—Sullivan Room-and-Pillar Chain Machine, Class CE-7.

TABLE I.—SULLIVAN AIR-DRIVEN CHAIN MACHINES

	CLA 2	CE 7
Length of machine body, inches	93¾	90
Width, inches	30	32
Height, inches	12*	24½
Height of undercut (kerf), inches	5 or 7	5¾
Length of chain-bar (jib), inches	30 to 90†	66 to 90
Net weight (less jib), pounds	4,075	5,800
Weight of 4½-ft. jib, pounds	572	
Diameter of feed-rope drum, inches	16	(chain)
Drum capacity, feet of ½-in. rope	90	....
Speed of cutting feed, feet per minute	15 to 30	15 to 30
Air pressure, pounds	30 to 70	20 70
Horsepower of air motor (average)	30	30

* When cutting 7 in. kerf, height of machine is 14 in.

† Lengths of the standard jibs available vary by intervals of 6 in. from 30 to 90 in., with an 8-ft. special jib.

the face of coal, the frame or jib carrying the cutter chain lies close to the floor, the main sprocket being then underneath the driving gearing. In another form of the machine (Fig. 8), for making an "overcut" 2½ to 3 ft. above the floor of the seam, the sprocket is placed above the driving gear. The same illustration also shows how the entire machine can be raised above its bedplate by four jack-screws, to make an overcut in any part (within limits) of the face of coal.

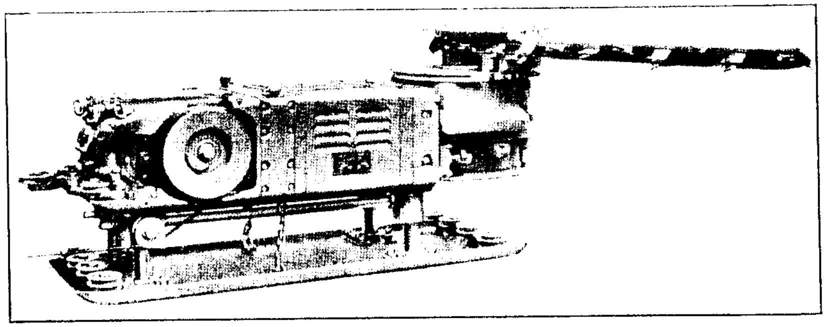


FIG. 8.—Mavor & Coulson's Air-driven Chain Machine (with set-up for "overcutting").

This coal cutter, widely used in Great Britain and elsewhere, is made in two sizes (Table II). It will be seen from its dimensions that it will work in very thin seams.

Rotary-bar Cutters were formerly used to a limited extent in the United States for room mining, but are no longer made here. In Europe, also, they have been largely superseded by chain machines.

TABLE II.—MAVOR & COULSON'S AIR-DRIVEN CHAIN CUTTER

	Model AXN	Model BXN
Length, inches	98½	108
Height, inches	15	19¾
Width, inches	30½	33¼
Net weight, pounds	4,800	6,770
Air pressure at machine, pounds	45	45
Cubic feet free air per minute, normal pressure, full load	400	600
Horsepower at normal pressure	25	35
Depth of undercut, inches	42 to 54	42 to 72

Maximum depth of cut depends upon the hardness of the coal or other material.

Fig. 9 shows the operating parts, with the casing removed, of the latest form of bar cutter for longwall mining, built by Mavor & Coulson. It is driven by an air turbine, similar in principle to that in Figs. 4 and 8. There are three sizes, the specifications of which are given in Table III. In this improved design, the turbine rotors are provided with axial and radial adjustments to maintain proper clearance and to take up wear.

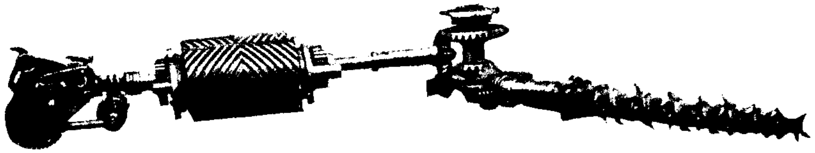


FIG. 9.—Operating Parts of Mavor & Coulson's Bar Cutter.

The tapering cutter bar has a series of staggered bits, set in sockets throughout its length, and is threaded spirally, to draw out the cuttings from the groove as the machine advances along the face of coal. The bits are readily removed for sharpening and replacement.

Like other coal cutters, this machine may be mounted on runners or skids for sliding on the floor of the seam, and can be moved on a track from place to place by means of a low, light truck. To groove a seam at a distance above the floor, the cutter is stood on a base-frame, which raises it sufficiently to make the cut at a height of say 4 or 5 ft.

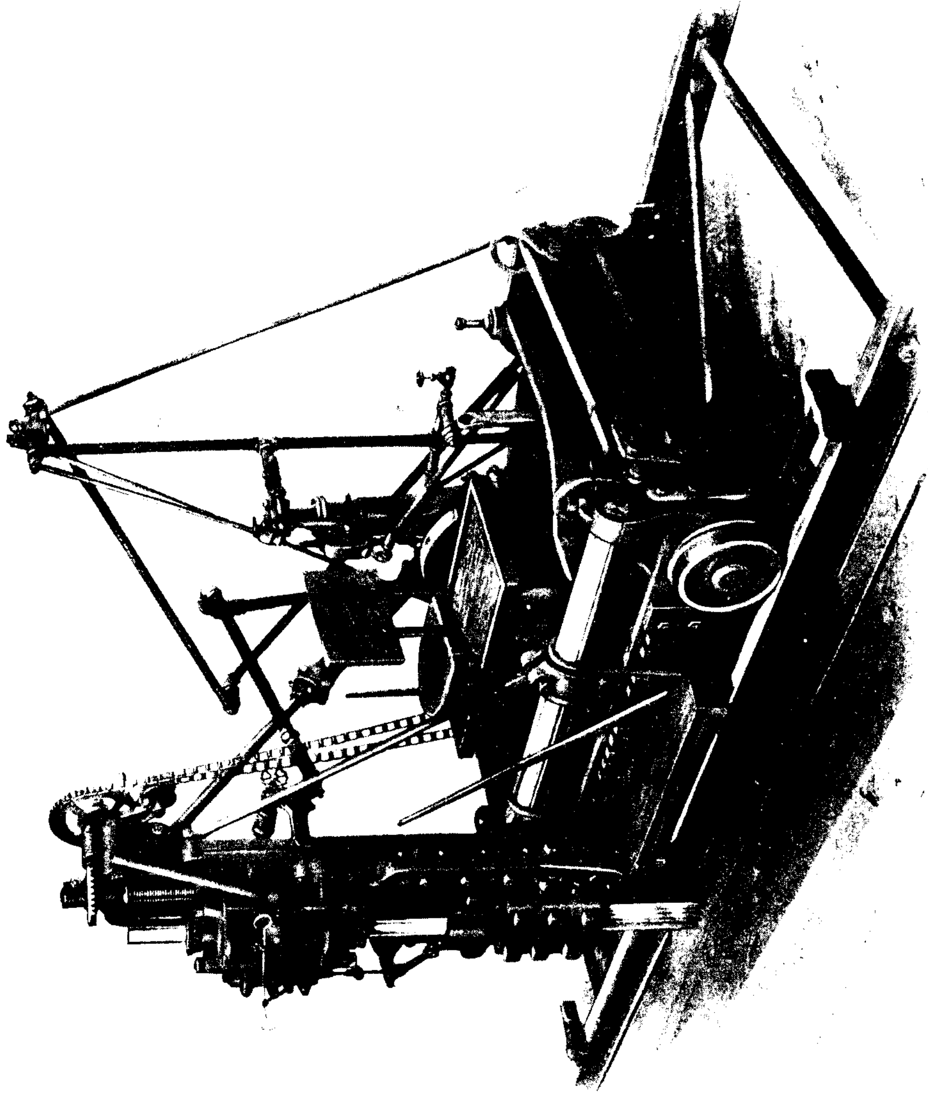


FIG. 19.—Sullivan Track Channeler, Type VX (New Albany Machine Co., New Albany, Ind.

gang of bits is set vertically (Figs. 19 and 20). For channeling in inclined strata, certain adjustments are necessary, for setting the cylinder in the proper angular position. These adjustments are of two kinds: *first*, the standard and cylinder can be canted (tilted) backward over the carriage, and, *second*, the standard can be swung in a plane parallel to the line of the track on which the machine stands. For "double-angle" work, both adjustments are made simultaneously (Fig. 21).* In different designs of channeler, the "swing-back" angle (across the track) varies between limits of say 5° and 45° to the vertical; the maximum angle parallel to the track is usually 15°.

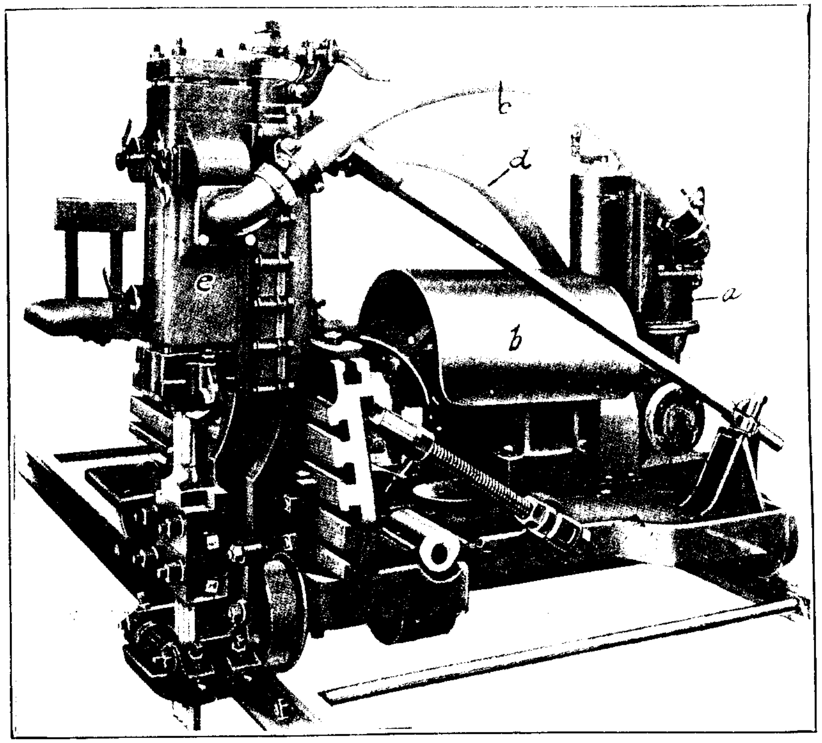


FIG. 20.—Ingersoll-Rand "Electric-air" Track Channeler.

In Fig. 19 the standard carrying the cylinder (corresponding to the cradle of a rock drill) is attached to a collar on the longitudinal trunnion bar, along which the standard can be moved and bolted in the desired position. Behind the standard is a long, threaded, inclined rod, by which the cylinder head is canted backward (first angular adjustment).

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* This cut, of an older design of Sullivan Channeler, shows the position of the cylinder with both angular adjustments.

By another inclined rod, the upper end of which is pivoted near the top of the standard, and the lower end attached to an adjustable collar on the trunnion bar, the cylinder is swung to the required angular position parallel to the track (second adjustment).

In the Ingersoll-Rand channeler, Fig. 20, the standard and cylinder are carried by, and can be slid longitudinally along, a heavy cast plate, having T-slots like the table of a machine-shop planer. Behind the right-hand end of the plate is shown one of a pair of inclined, threaded rods, by which the plate and cylinder are canted backward. By these means all the necessary adjustments are made. This is not a true compressed-air machine. An air "pulsator" *a* is driven by a direct- or alternating-current motor *b*. The ends of the pulsator cylinder are

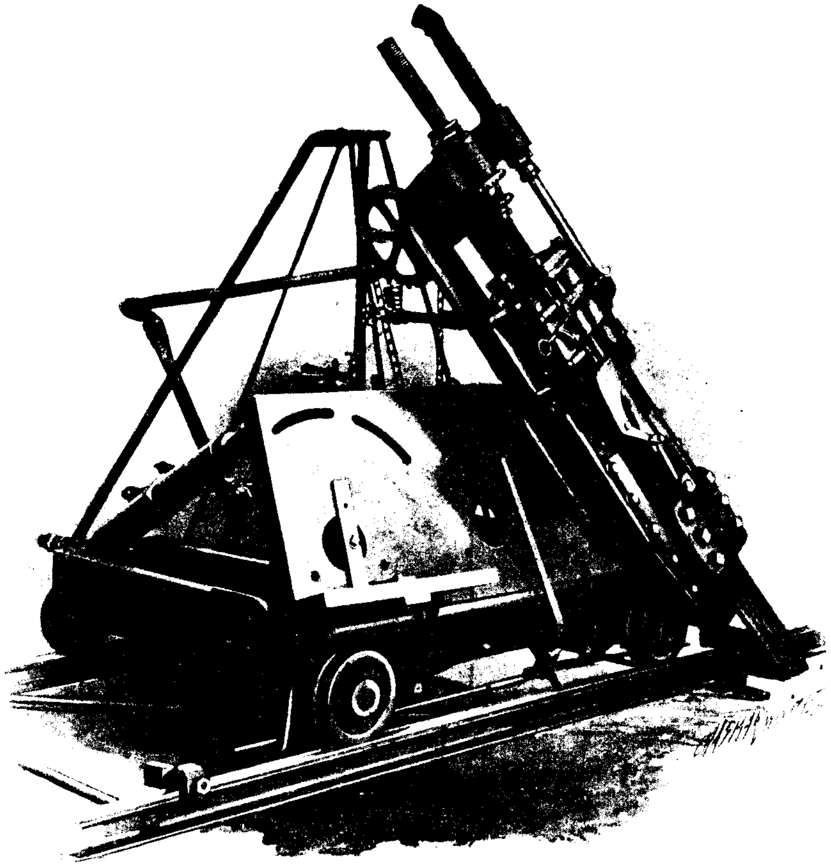

Fig. 21. Sullivan Adjustable-back Channeler (New Albany Machine Co.)

connected by hose *c* and *d* with the channeler cylinder *e*, thus causing the reciprocations of the gang of bits. For specifications, see Table VI. In principle this machine resembles the former Gibson-Ingersoll channeler, the design of which in turn was based on the obsolete Temple-Ingersoll electric-air drill.

Fig. 22 shows a Sullivan machine * (similar to that in Fig. 19), but arranged for horizontal channeling or "undercutting." The change for this purpose is readily made by the use of a few extra parts, the standard and cylinder being slung under the longitudinal trunnion bar, at either end of the channeler.

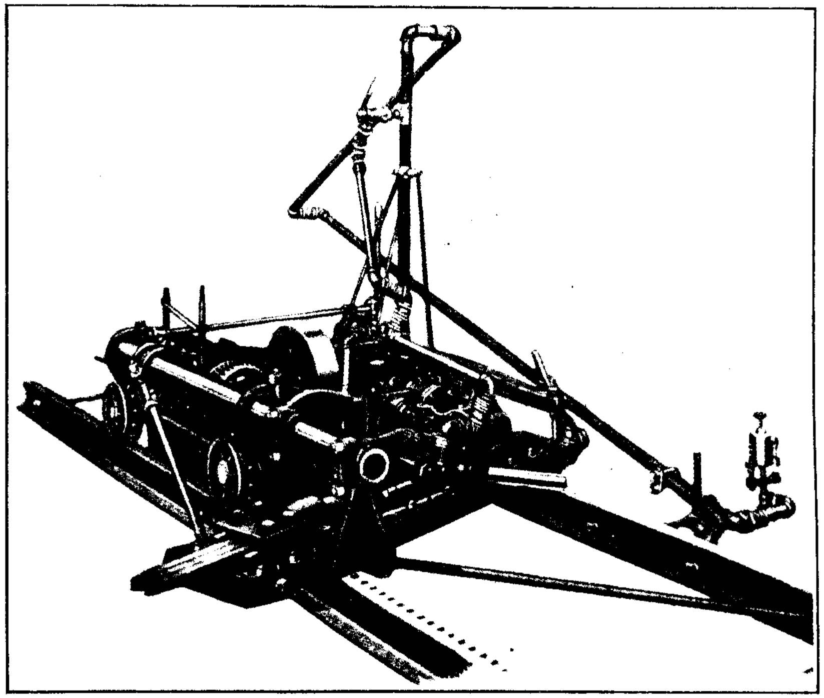

Fig. 22.—Sullivan Channeler, Type VX 31, Arranged for Undercutting (New Albany Machine Co.). Length, 51 in.; width, 54 in.; track gage, 39 in.; net weight, 2,950 lb.)

One of the Sullivan channelers is a duplex machine, with two cylinders mounted side by side on a heavy standard; each cylinder has its gang of bits, and both are operated by a single valve-chest. As the up and down strokes alternate, high speeds are possible without excessive vibration.

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* The Sullivan and Wardwell channelers are now built by the New Albany Machine Co., New Albany, Indiana.

The working capacity is correspondingly greater than that of the single cylinder machines.

Tables V and VI, containing specifications of air-driven channelers, will further illustrate the features of these machines.

Among other channelers are the Wardwell* and the Bryant. The former is a heavy steam- or electric-driven machine, designed for vertical channeling only, as in various kinds of engineering excavation.

Another type of channeling machine, differing entirely from the preceding, consists of a large rock-drill provided with a swivel-plate for making the angular adjustments, and mounted on a pair of bars supported at each end by legs. The drill is fed along the bars by a small, 3-cylinder engine, which through gearing actuates a traveling feed-nut on a threaded spindle between the bars. The bit is of the usual gang type.

TABLE V. SULLIVAN AIR-DRIVEN CHANNELERS (New Albany Machine Co.)

	VW 22	VS 32	VS-72	VW 62
Length overall, inches	81	81	81	81
Width, overall, inches	101$\frac{3}{8}$	100$\frac{3}{8}$	100$\frac{3}{8}$	105$\frac{1}{2}$
Height, overall, inches	82	105	114	150
Center of cut to wall, inches	8	7	8	11
C. to c. of cuts, machine turned on track, inches	102	100	102	102
Maximum swing-back angle of standard from vertical, regular brace, degrees	30	40	40	5
Bore of main cylinder, inches	6$\frac{1}{2}$	7	8	8
Maximum stroke, inches	7$\frac{5}{8}$	8$\frac{13}{16}$	8$\frac{7}{8}$	10
Maximum feed run on standard, inches	20	20	26	40
Length of starting gang, inches	34	34	40	30
Differences between gang lengths, inches	18	18	24	36
Size of steel, 3-piece gang, inches	1 × 1$\frac{1}{4}$	1$\frac{1}{8}$ × 1$\frac{3}{8}$		1$\frac{1}{8}$ × 1$\frac{1}{2}$
Weight, machine only, pounds	13,400	11,950	12,300	15,340
Weight, equipment, pounds	4,900			5,370

NOTE. Sullivan channelers are built in thirteen types or sizes, of which the four in the table are air-driven, and equipped with reheaters.

Blocks of stone may also be quarried by an ordinary rock-drill, mounted on a single bar supported by adjustable legs. A row of closely spaced round holes is drilled, and the partitions between them are subsequently cut out by a "broaching" bit. Or, the holes may be blasted by light charges, or the block of stone loosened by wedges.

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* The Sullivan and Wardwell channelers are now built by the New Albany Machine Co., New Albany, Indiana.

TABLE VI. INGERSOLL-RAND "ELECTRIC-AIR" CHANNELER

Length, overall, inches	63	Stroke, inches	7
Width, overall, inches	96	Weight, including motor and rheostat, pounds	8,200
Height, overall, inches	75$\frac{1}{2}$	Shipping weight, with track and equipment, except gang steels, pounds	13,250
Center of cut to wall, inches	7$\frac{1}{2}$	Voltage, a.-c. or d.-c.	220
C. to c. of cuts, machine reversed, inches	89	Horsepower	10–15
Track gage, inches	70$\frac{1}{4}$		
Cylinder diameter, inches	7		

Channeler Bits of standard type consist of a "gang" of three to five separate, square-shanked units, set closely side by side, to insure a channel with smooth walls. There are several different designs: (*a*) Fig. 23 shows a five-bit gang, as generally used for quarrying marble. The cutting edges are alternately at right angles and at 30° to 45° to the direction of the channel, thus forming practically a succession of Z-shaped bits. (*b*) For tough stone, the normal and inclined edges of the adjacent units do not form closed Z's, as in Fig. 23, but are separated, the inclined cutting edges being at 45° to 60° to the line of the channel. (*c*) For open-cut engineering excavation, large single Z bits are often used instead of a gang, the inclined edge being at say 30° to the line of the channel. In starting a channel, the width or gage of bit is from 1$\frac{1}{2}$ to 4 in., depending on the depth of cut. The width is reduced in the successive gang lengths by $\frac{1}{16}$ in., finishing bits being say 1$\frac{3}{8}$ or 1$\frac{1}{4}$ in. The gang is clamped in a chuck on the piston rod of the cylinder, and is guided by a crosshead or a pair of roller guides. Speed of stroke, 300–400 per minute.

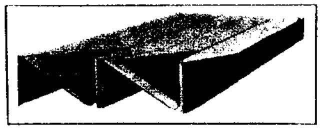
FIG. 23.—Five-bit Gang for Channeling Marble.

Performance of Channelers.—The depth of cut, with from two to four sets of bits, rarely exceeds 8 or 10 ft., though cuts to 18 or even 20 ft. can be made. No general statements as to capacity or speed of cutting are possible. It is greatest in soft stone of uniform texture, and where the work is systematic, as in large quarries. Examples of daily averages: in hard marble, 50–65 sq. ft.; softer marbles, 125 to 150 ft.; medium limestones, up to 200 ft.; soft sandstone, 200 to 250 ft.; slate, 60 to 75 ft.; hard gneiss, say 70 ft. Duplex channelers will make nearly double the above averages.

CHAPTER XXIII

PUMPING BY COMPRESSED AIR*

PUMPS operated by compressed air are of three kinds:

1. Direct-acting pumps, driven by compressed air instead of steam.
2. Pneumatic displacement pumps.
3. Air-lift pumps.

DIRECT-ACTING PUMPS

These pumps, primarily designed for steam, may also be operated by compressed air. Although of rugged construction and reliable in operation, they are always uneconomical when of the ordinary single-cylinder or duplex types. But, being readily obtainable in standard sizes, they are sometimes employed for mine service as a matter of convenience. Their features will be outlined briefly.

As pumps work under practically constant load, which is the weight of water in the delivery pipe plus friction, and as they seldom have fly-wheels to store up energy, the amount of expansion of either steam or compressed air, corresponding to the point of cutoff, must be relatively smaller than that which would be suitable in ordinary stationary engines. Moreover, in the absence of a flywheel, the actual length of stroke of a pump is controlled only by its valve motion, in admitting and exhausting the steam or air. Simple, single-cylinder pumps usually work with little or no cutoff, so that a cylinderful of steam or air, nearly at initial pressure, is exhausted at each stroke. Duplex pumps, having a positive, interdependent valve motion, may at times operate even more uneconomically than those with a single cylinder; because, due to wear of parts, one piston may reach the end of its stroke before the other is in position to reverse its valve. Working conditions are considerably improved by using compound or triple-expansion pumps. These are suitable for heavy duty, in raising large volumes of water, and may then be provided with a flywheel. If driven by compressed air, the

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* The reader is referred for further details to a useful text-book, "Pumping by Compressed Air," by E. M. Ivens, published in 1914 by John Wiley & Sons.

exhaust from the high-pressure cylinder should be reheated before passing to the low-pressure cylinder, to restore the heat lost by expansion.

The fundamental differences in the behavior of steam and compressed air, when used in reciprocating engines and pumps, were discussed in Chapter XVI. These differences are chiefly due to the greater drop in temperature of compressed air as it expands in the cylinder; whence it follows that the position of the compressed-air expansion curve, in an indicator diagram, is lower than that of the steam curve. This means lower mean and terminal pressures, and the power developed is correspondingly diminished.

Notwithstanding the unfavorable working conditions set forth above, when reciprocating pumps are to be driven by compressed air for mine service it is customary to select from makers' lists standard pumps as designed for steam. Alterations in their valves and the proportions of their cylinders, to adapt them to the existing conditions, are rarely made. The need for such changes is evident. In a mine having a number of levels the lowermost pump must work under a heavier head than the others. As all are usually supplied with air at the same pressure, by branches from the main air line, the differences in the heads should be met by properly proportioning the diameters of the power and water cylinders of each pump. If, through error or neglect this is not done, the deepest pump in the mine may be the only one operating with a proper air pressure. The power cylinders of the others may require a pressure of only 30 or 40 lb., while the line pressure is 80 lb. or more, as is usual with compressor plants supplying air for rock-drills; their cylinders, therefore, even when the pumps are running under throttle, will be filled with air at nearly full pressure at the moment of exhaust.

To sum up, the adverse conditions under which direct-acting pumps generally operate are: lack of a flywheel, non-expansive use of the air, and unsuitable cylinder proportions. They are inevitably uneconomical, whether driven by steam or compressed air.

There are times, however, when compressed-air drive may be adopted with advantage for mine pumps. Electric drive is distinctly economical and is widely used, but electric current is not always locally available. Steam for underground service has three disadvantages: the heavy condensation loss in long pipe lines; heating of the mine atmosphere, due to radiation from piping and pumps; and the difficulty of disposing of the exhaust steam. By using compressed air these difficulties disappear. Compressed air can be conveyed long distances with small loss (see Chapter XV), the exhaust air somewhat improves the ventilation of the mine workings, and, in deep mines, if the piping is of sufficient diameter to minimize friction, the air pressure may be actually increased, due to the weight of the column of air in the main.

Simple, Single-cylinder Pumps.—The size of the water cylinder is first determined. There are three variables to be dealt with: diameter, length of stroke, and number of strokes per minute; or the last two factors may be combined as piston speed. For a given air pressure and head of water, the diameter of the air cylinder depends upon that of the water cylinder. This relation is conveniently based upon the assumption that a column of water 1 in. square by 12 in. high weighs 0.5 lb. (actual weight = 0.434 lb.); then, area of air cylinder is to area of water cylinder as half the head is to the air pressure.

In Table I the unit diameter of water cylinder is taken as 1 in. Diameters of air cylinders, as calculated, will be in decimals, and often of odd sizes. After finding the exact diameter, the nearest standard size is chosen from makers' lists, and the air pressure and piston speed adjusted accordingly.

Table I.—Ratios of Diameter of Air Cylinder to Diameter of Water Cylinder (William Cox)

Head in feet	Air Pressure, Pounds						
	20	25	30	35	40	45	50
50	1.12	1.00	0.91	0.84	0.79	0.74	0.71
100	1.58	1.41	1.29	1.20	1.12	1.05	1.00
125	1.77	1.58	1.45	1.34	1.25	1.18	1.12
150	1.94	1.73	1.58	1.45	1.37	1.29	1.22
175	2.09	1.87	1.70	1.58	1.48	1.39	1.32
200	2.24	2.00	1.82	1.69	1.58	1.49	1.41
225	2.37	2.12	1.94	1.79	1.68	1.58	1.50
250	2.50	2.24	2.05	1.90	1.77	1.67	1.58
275	2.62	2.35	2.14	1.98	1.85	1.75	1.66
300	2.74	2.45	2.24	2.07	1.94	1.82	1.73
325	2.85	2.55	2.33	2.16	2.02	1.90	1.80
350	2.96	2.64	2.42	2.24	2.09	1.97	1.87
375	3.06	2.74	2.50	2.31	2.16	2.04	1.94
400	3.16	2.83	2.58	2.39	2.23	2.11	2.00
425	3.26	2.92	2.66	2.46	2.30	2.17	2.06
450	3.35	3.00	2.74	2.53	2.37	2.24	2.12
475	3.44	3.08	2.82	2.60	2.44	2.30	2.18
500	3.53	3.16	2.89	2.67	2.50	2.36	2.24

For a single-cylinder pump,

$$V = 0.093W_2\frac{h \times G}{P},$$

in which V = volume of air in cubic feet per minute;
h = head of water, feet;
G = water raised per minute, gallons;
P = receiver gage pressure of air to be used, pounds;
W_2 = volume of free air corresponding to 1 cu. ft. at pressure P.

In this formula (by Cox), 15 per cent has been added to the volume of air to cover losses. Table II gives values of $0.093\ W_2$ for different pressures:

Table II

Air pressure, P, pounds	$0.093\ W_2$	Air pressure, P, pounds	$0.093\ W_2$
30	0.28272	65	0.50406
35	0.31434	70	0.53568
40	0.34596	75	0.56730
45	0.37758	80	0.59890
50	0.40920	85	0.63054
55	0.44082	90	0.66216
60	0.47244		

Table III gives unit horsepowers and mean pressures for different terminal pressures. As the horsepower corresponding to a given terminal pressure does not increase in constant ratio with the initial air pressure, it follows that the higher pressures are less economical for simple pumps.

Table III

Terminal pressure, pounds	Mean pressure per stroke, pounds	Horsepower per cubic foot free air
30	19.40	0.0847
35	21.60	0.0943
40	23.66	0.1033
45	25.59	0.1117
50	27.39	0.1196
55	29.11	0.1270
60	30.75	0.1340
65	32.32	0.1406
70	33.83	0.1468
75	35.27	0.1527
80	36.64	0.1583

But the low pressures suitable for simple pumps will not serve for machine drills, which are always the chief users of compressed air in mines and quarries. As a separate, low-pressure compressor would generally be out of the question, the high-pressure air must either be wire-drawn into the pump cylinder or reduced to the required pressure by a reducing valve.

In the first case, volumes of air already computed must be increased by a factor based on the ratio which the pressure for the pump bears to the pressure in the air main. Table IV shows the volumes of free air theoretically required for a unit of 10,000 ft.-gal. of work (=83,000 ft.-lb. or 2.5 hp.), at different air pressures, referred to a standard receiver pressure of 90 lb.

TABLE IV.—(EDWARD A. RIX)

Gage pressure, lb.	Ratio of compression, referred to 90 lb.	Cu.ft. of air calculated from Cox's formula	Factor of increase for wire-drawing from 90 lb.	Increased volume, cu.ft.	Actual hp. at 90 lb.	Efficiency on basis of 2.5 hp. theoretical
20	3.0	113	1.26	142	28.6	9.0
25	2.6	108	1.22	125	25.0	10.0
30	2.3	97	1.19	115	23.0	11.0
35	2.1	93	1.17	108	21.5	11.6
40	1.9	89	1.14	102	20.5	12.2
45	1.7	87	1.12	97	19.7	12.7
50	1.6	85	1.11	93	19.0	13.1
55	1.5	82	1.09	89	18.2	13.7
60	1.4	80	1.07	86	17.1	14.3
65	1.31	79	1.06	84	16.8	14.9
70	1.24	78	1.05	82	16.4	15.3
75	1.17	77	1.04	80	16.0	15.6
80	1.1	76	1.03	78	15.6	16.0
85	1.05	75	1.02	76	15.2	16.4
90	1.0	74	1.0	74	14.8	16.9

The factors in column 4 are assumed as 70 per cent of the ratios of the absolute temperatures due to expansion of the air from 90 lb., to the pressures in column 1. They will apply when the length of air main from compressor pump is moderate, as for a pump situated at the bottom of a shaft.

In the second case, the normal air pressure in the mine is reduced by an adjustable, spring reducing valve in the air main. The resulting increase of volume is accompanied by a drop in temperature, so that the full increase is not realized. Part of the lost heat will be regained from external sources if there is any considerable length of pipe between the reducing valve and pump. The efficiency will be further increased if the partly expanded air is passed through an underground receiver to the pump (see Chapter IX). The air will thus regain nearly its normal temperature, much of the entrained moisture will be deposited and trouble from freezing avoided, and the receiver, if near the pump, will minimize the air pulsations. By reheating, the efficiency can be increased considerably beyond the efficiencies shown in Table IV.

The prevention of freezing of moisture has been discussed in Chapter XVII. In connection with pumping, some benefit may be derived by leading a jet of water from the pump column into the air pipe, close to the pump. A very small quantity of water will prevent excessive drop in temperature of the exhaust. A better way is to tap a $\frac{1}{4}$-in. pipe into the column pipe, draw down its end to, say, $\frac{1}{32}$ inch and insert into the exhaust port the nozzle so formed. If water at 212° F. is injected, each pound in cooling to 32° F. gives up 180 thermal units.

Air-driven Compound Pumps.—It is commonly assumed that in compound pumps compressed air should be employed like steam, with a cutoff in each cylinder. The drop in cylinder temperature would obviously be less than that caused in a single cylinder by the same ratio of expansion from a given initial pressure. But in aiming thus to attain a higher efficiency, by adopting a large range of expansion, the cylinder temperature would still be very low. The loss of heat takes place chiefly within the cylinder, instead of at the exhaust port, as with pumps working at full pressure. Furthermore, the low terminal pressure caused by an early cutoff decreases the force of the exhaust and the ports are more liable to be choked with frost.

When using air expansively reheating is necessary, aside from any question of gain in economy (see Chapter XVIII). There are several methods: (1) preheating the air sufficiently to permit a reasonably early cutoff in each cylinder, while still avoiding too low a temperature in the low-pressure cylinder; (2) in addition to preheating, reheating the air between the cylinders; (3) using cold air at full pressure in the high-pressure cylinder and expanding into the low-pressure cylinder, with or without reheating; (4) using air at full pressure in both cylinders, the air being expanded between them in a receiver-reheater.

The first two methods are feasible when the pump is of suitable design and the heat properly applied; but there would be undesirable variations in power and speed if the pump has no flywheel. Under the first plan, a high initial temperature is necessary. If the expansion is adiabatic, from say 80 lb. to atmospheric pressure and normal temperature, the required preheating temperature would be:

$$T' = T\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} \quad \text{or,} \quad T' = 70^\circ + 459^\circ\left(\frac{80+15}{15}\right)^{0.29} = 446^\circ \text{ F.}$$

Although this temperature decreases during the stroke, proper lubrication of the cylinder might still be difficult (see Chapter XVIII). With the third method there would still be a variable back-pressure on the high-pressure piston; also, the increase in volume due to clearances, and in the intermediate pipe, would considerably reduce the temperature of the air, and a large further drop would ensue during expansion in the low-pressure cylinder.

This temperature drop may be diminished by the fourth method, which is generally preferable to the others. Although only a small part of the total possible work of expansion within the cylinders can thus be realized, a compound pump so installed is much more efficient than a single-cylinder pump. Each air card is practically rectangular. The pressure drop between the cylinders needs to be only sufficient to cause an active flow of air into the reheater and thence to the low-pressure cylinder. A drop of 15 or 20 lb. for an initial pressure of 70–80 lb. will usually answer. The degree of reheating required to restore the heat lost by a drop of 20 lb., is only 204° F., for a final temperature of 60° at exhaust. If the pump is suitably situated, a fuel-burning or an electric reheater may be used; or, if this is inadmissible, the water from the suction or the column pipe may be utilized for warming the air, as already stated.

By employing reheating for properly designed and operated air-driven compound pumps, efficiencies of 40–50 per cent may be realized. Three cylinder pumps with intermediate heaters work at still higher efficiencies.

PNEUMATIC DISPLACEMENT PUMPS

In these pumps, and also the air-lift, compressed air acts directly on the water to be raised; there is no piston, cylinder, nor other moving part except the valves. Their maintenance cost is very low.

Pneumatic Displacement Pumps.—In these, the compressed air acts upon the surface of the water contained in a submerged tank, the admission of air and water being controlled by valves. The water displaced by the air is discharged through a pipe. There may be one or two tanks; with one the discharge is intermittent; with two it is practically constant. The working head corresponds to the air pressure employed. These pumps are especially useful for low heads and are advantageous for pumping acids, chemical solutions, etc., which would rapidly destroy a piston pump.

The double-chamber Merrill pneumatic pump is shown by Fig. 1. Compressed air enters through an automatic valve, which opens connection alternately with the water chambers. Since the pressure per square inch of a column of water is 0.434 lb. per foot of head, the height to which a given air pressure will raise water is equal to the gage pressure divided by 0.434; thus, air at 80 lb. will pump to a height of $\frac{80}{.434} = 184$ ft. To cover friction, leakage, absorption of air by the water, and to provide dynamic head for overcoming inertia and securing a proper speed of discharge, an added air pressure is necessary. One cubic foot of water is displaced per cubic foot of compressed air. One cubic foot of air at 80 lb. = $\frac{1 \times (80 + 15)}{15} = 6.33$ cu.ft. free air. Adding say 20 per cent for losses, the total is 7.6 cu.ft. free air per cubic foot of water.

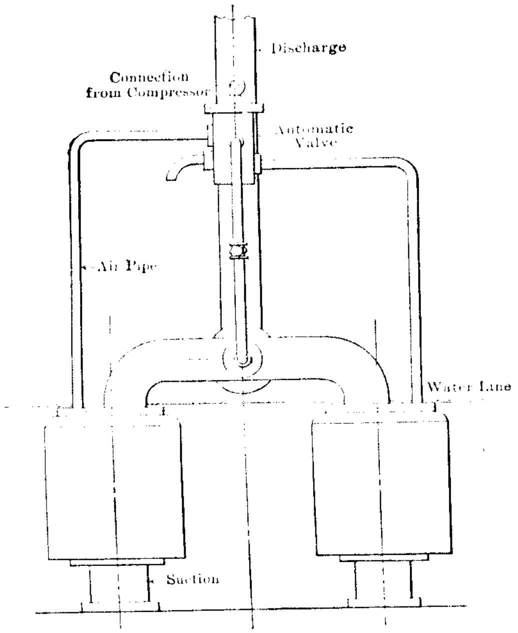

FIG. 1.—Merrill Pneumatic Displacement Pump (diagrammatic).

Example.—Since 1 gal. of water occupies 0.134 cu.ft., the work done per cubic foot of compressed air, against a head of 184 ft., will be: $\frac{184}{0.134 \times 7.6} = 180$ ft.-gal. = 1,503 ft.-lb. The actual work of compressing 1 cu.ft. of air to 80 lb. gage, by a single-stage compressor (Table II, Chapter X) is 0.183 hp., or 6,039 ft.-lb.; hence, the efficiency of the pump in the above case is nearly 25 per cent, which compares favorably with ordinary efficiencies of simple direct-acting pumps.

The displacement pump as in Fig. 1 exhausts at each stroke a tankful of air practically at gage pressure. By employing a series of pumps in a shaft, and using the air expansively, the height of lift for a given initial pressure, and the total efficiency, greatly exceed that shown above. With suitable valve control, the air is expanded from the lowermost tank to the one next above, and so on, the lifts decreasing toward the top of the series. When the last tank is discharged, the whole system is occupied by expanded air, at a pressure of 2 or 3 lb., which is then exhausted into the atmosphere. At 80 lb. air pressure, water can thus be raised to a height of about 330 ft., instead of 184 ft., as in the preceding example, and at an efficiency of about 40 per cent.

The Latta-Martin pump (Fig. 2), designed chiefly for raising large volumes of water under low heads, may be constructed for any desired air pressure and head. A pair of submerged tanks take water through large disk valves in the bottom. The air-valve mechanism, admitting air alternately into each tank, comprises a main and an auxiliary valve, like those of single-cylinder steam pumps. The auxiliary valve is thrown by the movements of two levers, from each of which is suspended a spherical bucket filled with water and hanging within the tank. As the tanks are alternately filled and emptied, the buckets rise and fall. For acid water, as in mines containing sulphide ore, the tanks may be lined with asphalt and the other parts made of bronze.

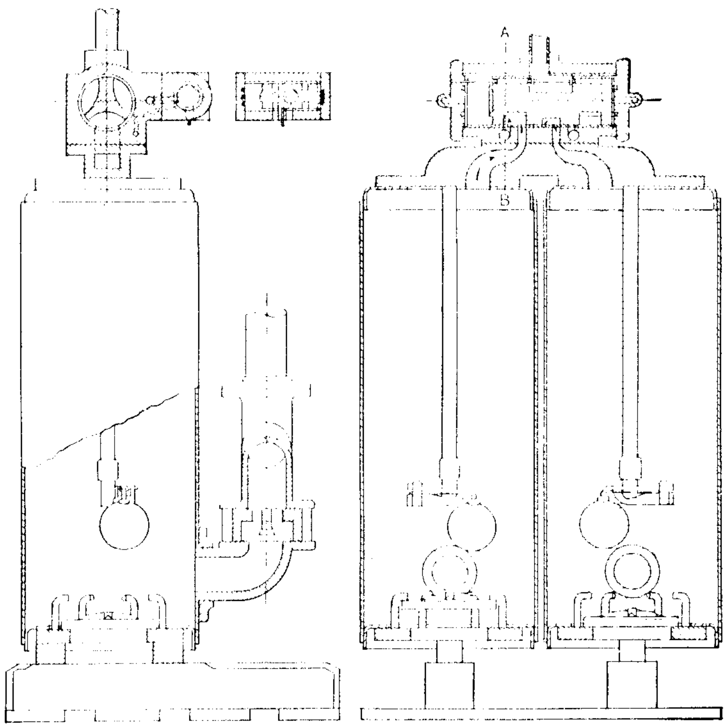

Fig. 2.—Latta-Martin Displacement Pump.

Return-air Displacement System uses compressed air expansively. Two tanks, either submerged or within suction distance of the sump, are connected by pipes with the compressor (if not submerged, the water enters by siphon action). Compressed air admitted to one tank forces out the water to the pipe-line, while the air is withdrawn from the other.

The charge of air is so regulated that, when one tank is empty, the other is full; then a switch valve reverses their action.

Referring to Fig. 3, *A* and *B* are the tanks, with their air pipes A^1, B^1; C^1 and C^2, discharge check valves, preventing return of the ejected fluid; D^1, D^2, check valves, preventing discharge through the inlets; *E*, the discharge pipe of both tanks; ***F***, an automatic switch controlling the pumping cycle; *J*, an automatic compensating valve, which takes care of loss and leakage. The exhaust from the tanks, instead of going to the atmosphere, is returned to the compressor intake. Hence, the system is a closed one, the same air being used over and over, as in the Cummings return-air compressor plant (Chapter XVI).

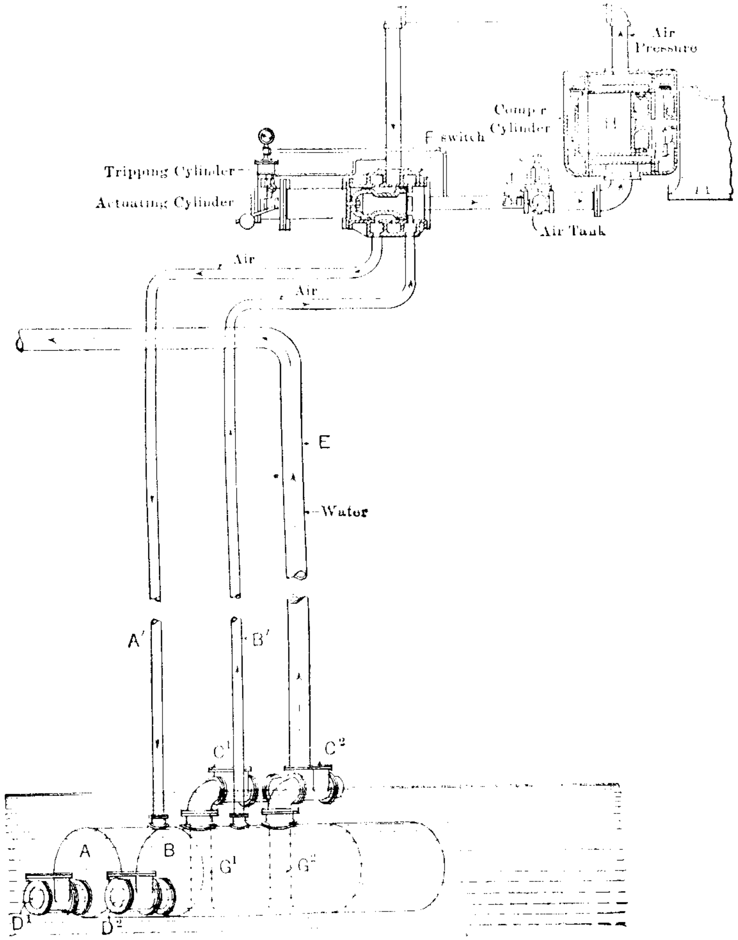

Fig. 3.—"Return-air" Displacement System (Ingersoll-Rand Co.)

In starting, after the water in one tank is expelled, the switch reverses, thus discharging the air from this tank to the compressor intake, where it acts expansively on the piston, reducing the power required to drive the compressor. Meanwhile, the second tank is discharging. When the pressure in the first tank has fallen sufficiently, it again fills with water. Valve *J* is set to open during the suction period, at a negative pressure a little greater than that required to draw water into the tanks. The switch-valve is operated by an electric make-and-break mechanism, controlled by a pressure gage on the air intake; or by a piston valve operated by a small air cylinder. The volume of air for a given size of tank may be determined in terms of revolutions of the compressor.*

This system has an efficiency of 55–60 per cent, and requires little attention during operation. It is now rarely used for pumping water; chiefly for elevating sands and slimes at concentration mills and for other similar service.

Table V.—Return-air System. Size of Compressor, Piping, Etc., for Different Heads. Tanks Fully Submerged

Lift, feet	Capacity of compressor, cu.ft. per min. piston displacement for 100 gal. per min.	Max. I.Hp. of air cylinder	Max. I.Hp. of steam cylinder	Average Hp. of steam cylinder	Area of air pipe, sq.in. for each 100 gal. capacity	Area of water pipe, sq.in. for each 100 gal. capacity
50	39.84	2.71	3.22	2.80	.96	7.70
60	42.78	3.28	3.85	3.37	1.03	8.25
70	45.30	3.85	4.53	3.93	1.09	8.73
80	47.70	4.45	5.22	4.49	1.14	9.12
90	49.80	5.03	5.91	5.05	1.20	9.60
100	51.84	5.67	6.67	5.61	1.25	10.00
120	55.41	6.96	8.18	6.73	1.33	10.60
150	59.94	9.00	10.60	8.41	1.44	11.50
170	62.64	10.42	12.25	9.54	1.50	12.00
200	66.12	12.76	15.00	11.22	1.58	12.65
250	71.10	16.46	19.35	14.02	1.71	13.70
300	75.06	20.45	24.00	16.83	1.80	14.40

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* For further details, see paper by Elmo G. Harris, *Trans.* Am. Soc. C. E., Vol. 54.

Air-lift Pump

Since 1888, when Dr. Julius Pohlé proposed the air-lift for pumping, it has become increasingly important. Besides raising water from wells, it is applicable also to pumping in shafts, and for elevating the slimes and sands of cyanide and concentration mills.

Principle and Operation.—The pump consists essentially of two pipes: a large delivery pipe, and a relatively small air pipe from the compressor receiver (Fig. 4). The delivery pipe, open at both ends, is submerged to a depth proportionate to, but normally greater than, the height to which the water is to be raised. The compressed-air pipe passes down to a point near the bottom, and admits air to the lower end or foot-piece of the delivery pipe. (Modifications of this arrangement are noted hereafter.)

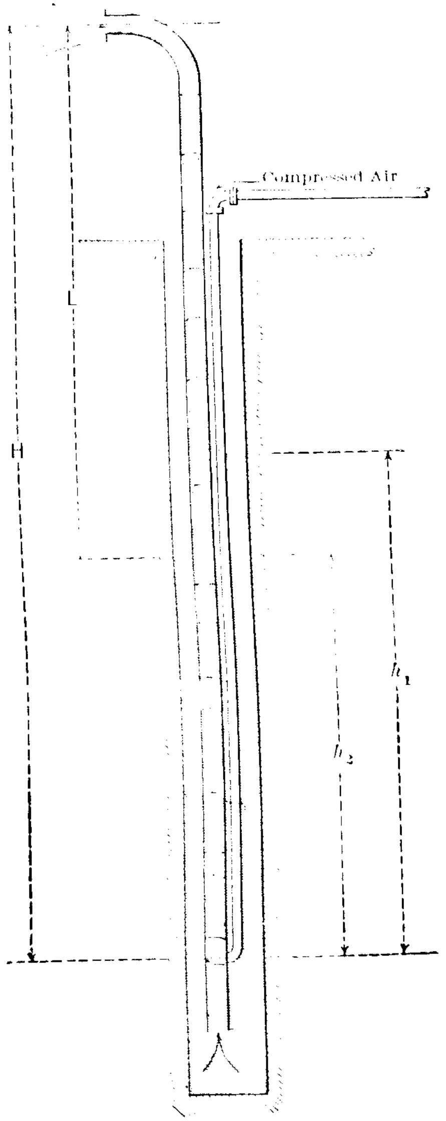

Fig. 4.—Diagram of Air-lift Pump.

In some respects the operation of the air-lift pump is the reverse in principle of the method of compressing air by the direct action of falling water (Chapter XIV). If the discharge pipe is of very small diameter, the compressed air entering the bottom tends to form piston-like layers, which rise rapidly, alternating with masses of water (as is shown by experimenting with glass tubes). If the discharge pipe is of large diameter, the air is best admitted through a series of ports or nozzles, resulting in a more complete dissemination of the air through the mass of rising water. The water is raised chiefly by the aeration of the column of water, which causes a reduction in its specific gravity; added to this is the expansive force and vis viva of the compressed air. Before the air is turned on, the water stands at the same level inside and outside of the delivery pipe. On entering the foot-piece, the air is under a pressure due to the weight of the rising column of water. As the bubbles of air rise, they expand with the decrease in head, and, on reaching the point of discharge, the tension of the air is reduced practically to atmospheric pressure. The initial air pressure required depends on the pressure due to head, measured from the point at which the air enters the delivery pipe to the surface of the water. At too high a pressure loss of work ensues at the compressor. Should the delivery pipe be too deeply submerged, in proportion to the net height of lift, an uneconomically high pressure will be required to force the air into the foot-piece; and, with insufficient submergence, more air will be necessary to produce the velocity of delivery.

Referring to Fig. 4,

let: h_1 = depth of the delivery-pipe foot-piece below the normal water-level, before pumping begins, or when the water is at rest;
h_2 = height of water-level when the pump is in operation;
H = height of the column of mixed air and water, measured from the air inlet to point of discharge;
L = net height of lift = $H - h_2$.

The compressed air enters the foot-piece at a pressure P', corresponding to the head h_2; or, $h_2 \times$ pressure per foot of hydraulic head $= 0.434\ h_2$. Assuming that the water rises in piston-like masses (as would be the case with a single air nozzle and a delivery pipe of small diameter), the sum of the length of these masses in the column H must theoretically be equal to the outside solid column of water h_2 (the weight of the compressed air contained in the column being neglected). But, to overcome the frictional resistance and produce flow, the head h_2 must be greater. Under ordinary working conditions, the net height of lift L is found to be from $0.5\ h_2$ to say $0.65\ h_2$. Taking the second value and transposing: $h_2 = \frac{L}{0.65}$; and by substituting in the expression for the value of P', as above: $P' = 0.434 \frac{L}{0.65} = 0.67\ L$. *If*, for example, L be 50 ft., $P' = 33.5$ lb., and $h_2 = \frac{50}{0.65} = 77$ ft.

Since the air in the column H is divided into small masses, surrounded by water, its expansion during the upward flow may be assumed to be isothermal. If P' is its initial pressure, the mean pressure for the entire lift $= P \times \text{Nap.log}\left(\frac{P'}{P}\right)$, P and P' being absolute pressures. In the above example, taking P as 15 lb., $P' = 33.5 + 15 = 48.5$ lb., and the mean pressure = 17.5 lb. gage.

For starting the pump, the air pressure must be sufficient to overcome the normal static head h_1, but, when flow has begun, the pressure required falls to that corresponding to h_2. This difference in pressure $(h_1 - h_2)$ is readily met by temporarily speeding up the compressor. To minimize fluctuations between h_1 and h_2, the upper part of the well or sump should have a large horizontal area, so that the water level is little affected by stoppages or variations in air pressure and delivery. The air throttle valve is regulated by a float on the surface of the water. The foot-piece must be properly designed, and the air pressure proportioned to the submergence and lift. Otherwise, air may leak back into the sump or outside the column of water; if the latter becomes aerated, more air will be required, thus reducing the efficiency.

For computing the performance of an air-lift Table VI may be used. The figures, which represent the work of well-proportioned plants, as to submergence and air pressure, show that the efficiency falls off rapidly as the air pressure and height of lift increase. Under normal conditions and with small lifts, efficiencies of 30 to 35 per cent are obtainable, and the efficiency may reach 45 or 50 per cent with proper air pressures and ratios of submergence to height of lift.

Volume of Air.—The following formula closely approximates average practice:

$$V_a = 0.8\left(\frac{h}{C \times \log \frac{H + 34}{34}}\right)$$

in which: V_a = vol. free air for raising 1 gal. water;
h = total lift, feet;
H = submergence when running, feet;
C = constant.

Values of C with proper submergence:

Lift in feet (h)	Constant	Lift in feet (h)	Constant
10– 60	245	501–650	185
61–200	233	651–750	156
201–500	216		

TABLE VI.—(E. E. JOHNSON)

Lift		Theoretical horsepower				Efficiency of Air-lift				
			To deliver 1 cu.ft. of air per minute			Theoretical			Total efficiency, from power applied to water delivered	
Pounds pressure	Feet, head	To lift 1 cu.ft. of water per minute	Isothermal	Two-stage	Adiabatic	Isothermal compression	Two-stage compression	Single-stage or adiabatic compression	Two-stage compression	Single-stage or adiabatic compression
5	11.54	0.02185	0.02514	0.02572	0.0263	0.87	0.848	0.83	0.623	0.497
10	23.09	0.04363	0.05586	0.05992	0.064	0.78	0.728	0.684	0.516	0.44
15	34.63	0.06545	0.09105	0.0962	0.1015	0.72	0.687	0.648	0.515	0.389
20	46.20	0.08727	0.12994	0.1394	0.1483	0.675	0.627	0.59	0.47	0.354
25	57.75	0.109	0.17191	0.1897	0.2004	0.635	0.575	0.545	0.432	0.327
30	69.31	0.13091	0.21678	0.2370	0.2573	0.603	0.548	0.508	0.412	0.305
35	80.86	0.1527	0.26445	0.2915	0.3187	0.577	0.52	0.478	0.39	0.287
40	92.41	0.17454	0.31375	0.3489	0.3842	0.557	0.502	0.455	0.376	0.273
45	103.90	0.1963	0.36368	0.4085	0.4535	0.540	0.482	0.433	0.362	0.260
50	115.50	0.21818	0.41848	0.4722	0.5261	0.522	0.464	0.415	0.348	0.249
55	127.00	0.24	0.47112	0.5366	0.6023	0.51	0.447	0.40	0.336	0.24
60	138.60	0.26181	0.52855	0.6051	0.6818	0.495	0.432	0.384	0.324	0.231
65	150.10	0.2836	0.58612	0.6734	0.7608	0.483	0.422	0.372	0.316	0.223
70	161.70	0.30545	0.64812	0.748	0.8483	0.471	0.408	0.36	0.307	0.216
75	173.30	0.3273	0.70952	0.823	0.9380	0.462	0.398	0.35	0.299	0.210
80	184.80	0.3491	0.76843	0.898	1.0291	0.455	0.39	0.343	0.292	0.206
85	196.30	0.37	0.83039	0.976	1.1241	0.446	0.38	0.331	0.285	0.198
90	207.90	0.3927	0.89444	1.055	1.2176	0.439	0.373	0.324	0.28	0.194
95	219.40	0.4145	0.96164	1.137	1.3148	0.434	0.368	0.315	0.276	0.189
100	230.90	0.43636	1.0243	1.247	1.4171	0.428	0.352	0.308	0.264	0.185
110	254.10	0.48	1.162	1.394	1.626	0.413	0.346	0.296	0.26	0.177
120	277.20	0.5236	1.301	1.571	1.841	0.402	0.333	0.285	0.25	0.171
130	300.40	0.5675	1.443	1.755	2.068	0.394	0.324	0.275	0.243	0.165

Air Pressure on starting is equal to the depth of the foot-piece in the well, less the water head in feet × 0.434. Working pressure equals the depth of foot-piece, less the pumping head × 0.434 + friction in air pipe + 2 lb. back pressure at the foot-piece.

Ratio of Lift to Submergence in average practice:

Height of lift, feet	Submergence, per cent	Height of lift, feet	Submergence, per cent
Up to 50	70–66	200–300	50–43
50–100	66–55	300–400	43–40
100–200	55–50	400–500	40–33

Size of Air and Delivery Pipes.—Ratios of these diameters are usually from 1 (air): 2 (water) to 1 : 2¼ or 3. The absolute diameters are determined on the basis of frictional loss caused by flow in the pipes. A water velocity of 250–300 ft. per min. may be assigned for the delivery pipe. For the friction losses in air pipes, see Chapter XV. When water is delivered at a distance from the pump, the added pipe resistance is determined, and air pressure and submergence correspondingly increased.

TABLE VII.—CALCULATED VALUES FOR AIR-LIFTS

Lift, feet	Volume of air per cu.ft. of water	Submergence, at 60% of total height of delivery pipe	Air pressure, pounds	Horsepower per gallon of water per minute
25	2.0	38	17	0.0181
50	3.0	75	33	0.0426
75	4.5	113	49	0.0828
100	6.0	150	65	0.1320
125	7.5	188	82	0.1910
150	9.0	225	98	0.2544
175	10.5	263	115	0.3150
200	12.0	300	130	0.3808

Foot-pieces.—Referring to Fig. 5, *A* is the original Pohlé side inlet, now rarely used. The foot-piece *B* (Ingersoll-Rand Co.) is a modification of *A*, admitting air around the periphery of the delivery pipe, without contracting the discharge area. It is furnished for capacities of 20–325 gal. per min. In the Saunders foot-piece *C*, for a cased well, the air passes down the annular space between the casing and delivery pipes; or, air enters a hollow base-plate studded with short vertical ¼-in. pipes. The water is thus split up and mixed with air on entering the delivery pipe. Design *D* is a variation of *C*. Design *E* (Sullivan Machinery Co.) and those in Fig. 6 (Ingersoll-Rand Co.) admit air through numerous small holes. *E* is a bronze casting; capacity, 5–1,500 gal. per min. In Fig. 6 the outer casing is steel, the inner tube, brass; capacities 15–2,000 gal. per min. To give an accelerated velocity, the three designs last mentioned have a Venturi contraction throat above the foot-piece.*

Tests were made in 1907 at the two 200-stamp mills of the Angelo and Cason mines, South Africa, on air-lifts for raising slimes and sands to the settling tanks. The delivery pipes were of two kinds: 10- to 16-in. pipes of constant diameter, and others increasing in diameter, from 12 and 14 in. at the bottom to 14 and 16 in. at the top. As a uniform taper was impracticable, the latter pipes, for a length of 35 ft. above the air inlet, were lined with 1 in. of wood, which served also to protect the metal from the scouring action of the sands.

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* For further details of air-lift apparatus see the catalogues of the Sullivan Machinery Co., and Ingersoll-Rand Co., which contain much useful information.

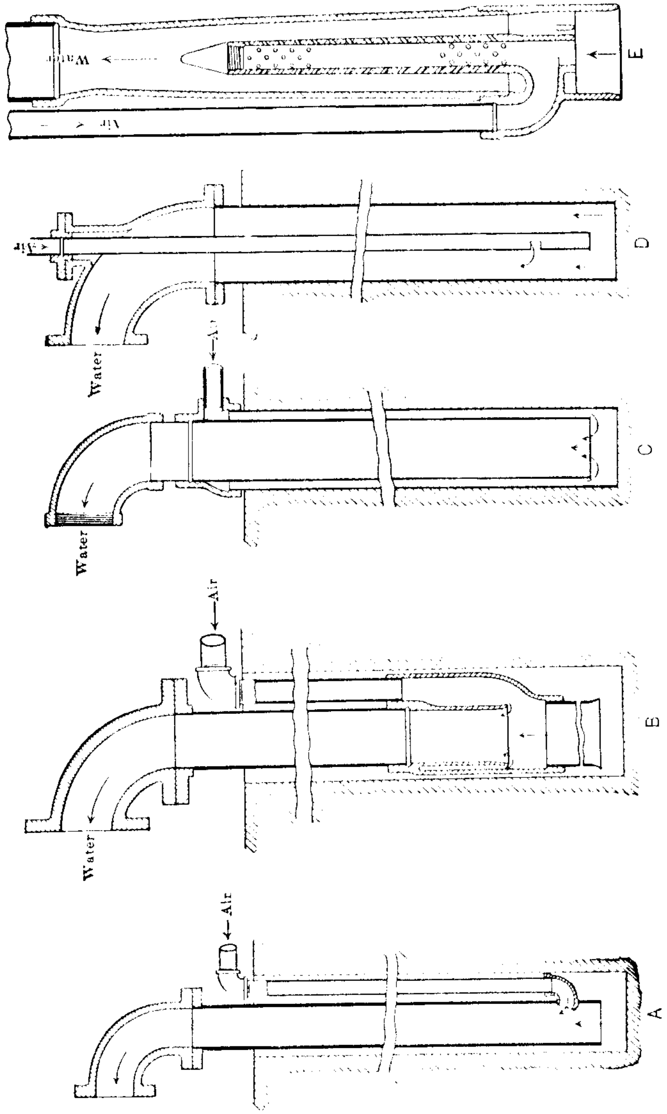

Fig. 5.—Foot-pieces and Modes of Piping for Air-lift Pumps.

The foot-piece used in the later tests was open at the bottom and

flared to double the diameter of the delivery pipe, to increase gradually the velocity of inflow (Fig. 7). A ring of 12 holes, 1 in. square, admitted the air. The foot-piece, supported on timbers, was open at the bottom for the free admission of the mixture to be pumped. The delivery pipe was of steel expanded into cast-iron flanges, the lower part lined with wood.

Table VIII presents details of four representative tests of the nineteen tests made. For computing the horsepower represented by the mixture of water and pulp raised, the weight of the slimes was found to be 63.3 lb., and of the sands, 64.56 lb., per cubic foot. Thus, for sands, the horsepower was:

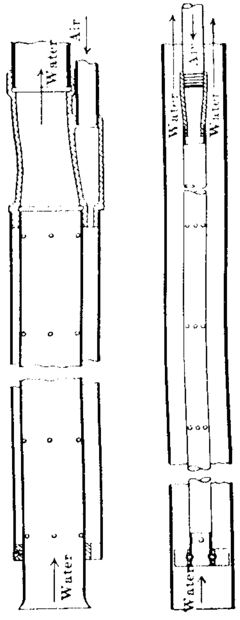

Fig. 6.—"VA" and "VC" Foot-pieces (Ingersoll-Rand Co.)

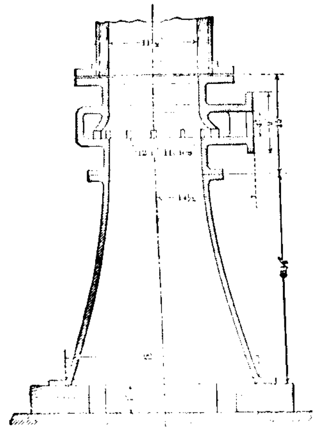

Fig. 7. Foot-piece for Air-lift for Raising Mill Tailings.

$$\frac{(\text{Quantity of sands} + \text{water}) \times 64.56 \times \text{ft. lift}}{33{,}000} = .001956 \times Q \times \text{ft. lift.}$$

The term "sands" refers to the mixture of water and crushed ore, from which the "slimes" have been separated.

Lansell's Air-lift is a modification of the air-lift, applied by Mr. George Lansell to pumping from a deep mine shaft in the Bendigo district, Victoria, Australia. A series of lifts were used for a depth of 1,385 ft. Fig. 8 shows diagrammatically two of the lifts.

TABLE VIII

Test	1	2	3	4
Conditions:				
Number and size of delivery pipes	Two 10-in.	Two 10-in.	One 16-in., decreasing to 14-in.	One 14-in., decreasing to 12-in.
Submersion in feet	32.75	35.75	37.75	48.85
Lift in feet	32.5	29.5	27.5	27.09
Ratio of submersion to lift	1.009:1	1.21:1	1.372:1	1.77:1
Gage pressure of air, pounds	15	16	17	22
Kind of foot-piece	Original	Original	Modified	Modified
Throat diameter of foot-piece	10 in.	10 in.	13½ in.	11½ in.
Performance:				
Free air, cubic feet per minute	2,256.0	1,279.0	746.48	846.0
Free air, per cubic feet of slimes	7.27	4.06	2.71	2.64
Cubic feet of slimes per minute	310.0	315.0	290.0	320.0
Throat velocity, cubic feet per second	4.7	4.8	4.85	7.39
Theoretical horsepower in pulp raised	19.3	17.8	15.23	16.6
Horsepower per cubic foot free air compressed	.048	.050	.053	.064
Air horsepower	108.72	64.71	42.21	54.14
Efficiency, per cent	17.7	27.5	37.15	30.55

The compressed air is conveyed from the receiver in a pipe *A*, running down the shaft. The water is conducted from a tank through a pipe *D*, which first passes down the shaft a certain distance, depending upon the height to which the water is to be raised, and is then connected with an enlarged section of pipe *E*, at the foot of the delivery pipe *B*. Thus, the piping for each lift forms an inverted siphon. At the lowest point of the siphon a short branch pipe *C* enters from the air main *A*, the end of this branch being directed upward into the foot-piece *E*. Before the compressed air is turned on, the water stands at the same level in pipes *D* and *B*. The effect of this arrangement is like that produced by submerging the lower part of the delivery pipe, as in the ordinary

air-lift. Check valves are placed, as shown, in pipes *D* and *C*, to prevent air or water from passing back into the air pipe or into the tank. A

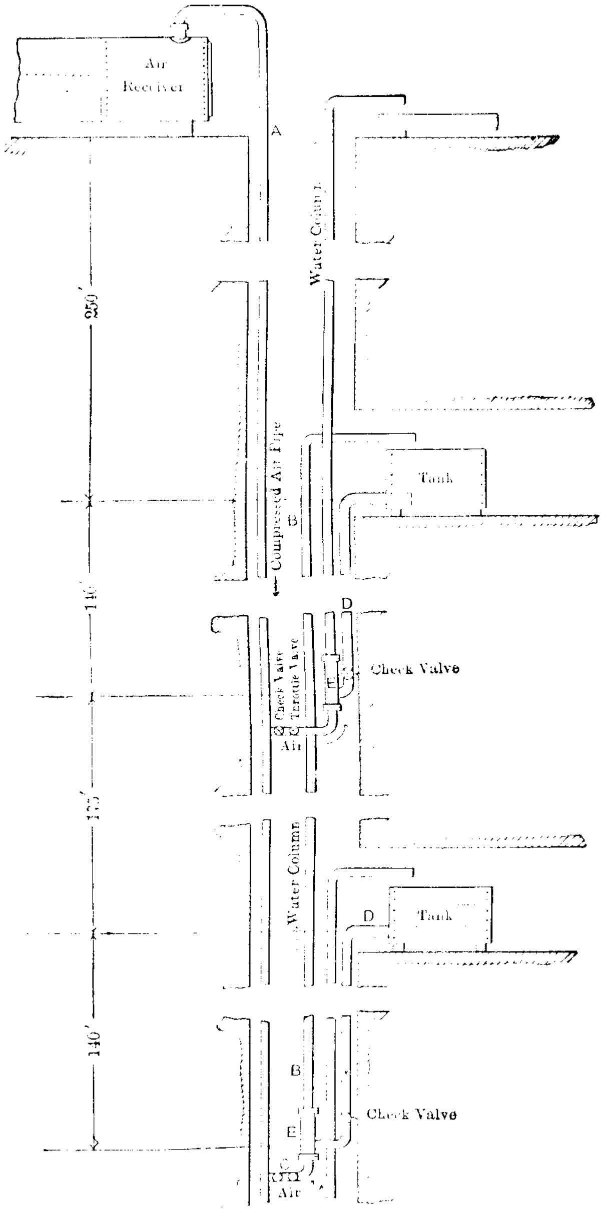

FIG. 8.—Diagram of Lansell's Air-lift Pump for Mine Shafts.

throttle valve in pipe *C* regulates the supply of air. The relative heights of the different parts are variable, the dimensions shown on the sketch indicating substantially the proper depth of the inverted siphon below the tanks, and the corresponding height of lift; thus, from

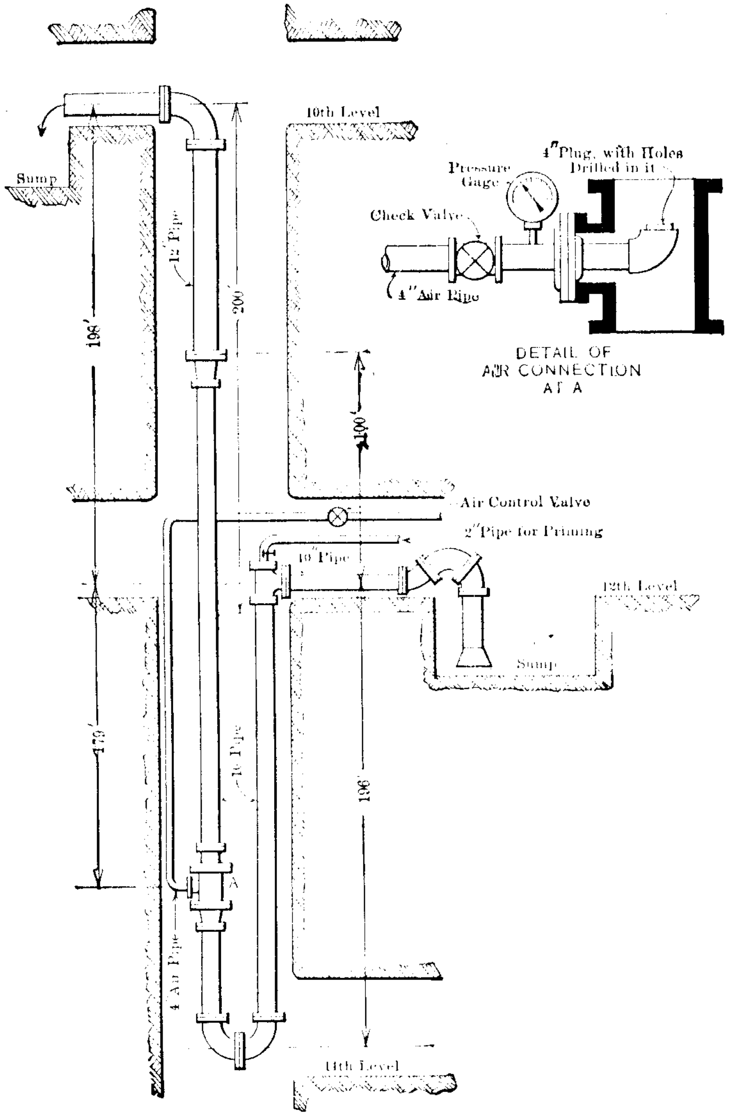

FIG. 9.—Air-lift at the Old Dominion Mine, Globe, Ariz. (*Trans.* American Institute Mining Engineers, Vol. 55).

the tank at the 250-ft. level, the pipe *D* passes down the shaft 140 ft., to the foot of the delivery pipe which discharges at the surface. By a series of lifts the water may thus be raised from any desired depth. The air pressure is the same for all, this pressure being 60–80 lb. per sq. in., or that which is ordinarily furnished for mine service.

Air-lifts at the Old Dominion Copper Mine, Arizona.*—Ten-inch air-lifts (Fig. 9) were installed from the 12th to the 10th level (200 ft.) and from the 10th to the drainage tunnel (431 ft.). The air lines were 4 in. The submergence on the lift from the 12th level was 177 ft., or 47 per cent; capacity, 1,650 gal. per min.; air pressure, 80–90 lb.; air consumption, 1,080 cu.ft. per 1,000 gal.; maximum efficiency, 36 per cent. Submergence on the 10th level lift was 188 ft., or 30.4 per cent; capacity, 1,233 gal. per min.; air pressure, 90–100 lb.; air consumption, 2,681 cu.ft. per 1,000 gal.; maximum efficiency, about 30 per cent.

During a period of unwatering the 16th level of the mine, the columns of two direct-acting pumps were converted into air-lifts.

Two-stage Air-lift at Burbanks Main-Lode mine, near Coolgardie, Western Australia, is used for raising and transporting mill sands (Fig. 10). The height of lift for single-stage would have required sinking a pit 39½ ft. deep, for the necessary submergence; for two-stage, the pit was only 15½ ft. Above the pit are two wooden head-boxes, with water levels 9 ft. apart. The wells for the pulp are of 6-in. cast-iron pipe; inside pipes are 3 in., and the air pipes ¾ in. in diameter. The air outlets of the latter are slots, at an angle of 45°, thus giving the air jets an upward direction. Supply of air is regulated by floats, so that a nearly constant level is maintained in each head-box, and no air is blown to waste. As the sand is heavy, the pulp mixture is 2½ water to 1 sand. Results obtained are as follows: free air per minute, 47 cu.ft.; sand per minute, 320 lb.; water per minute, 800 lb.; sand per cubic foot of free air, 6.8 lb.; sand and water per cubic foot of free air, 23.83 lb.

An air-lift for raising and discharging tailings is also used at the concentrating mill of the Chino Copper Co., New Mexico (*Compressed Air Magazine*, Jan., 1922).

Air-lifts at the Tiro General Mine, Charcas, Mexico. In 1919–20 this mine, about 1,300 ft. deep, was unwatered by air-lifts designed by S. F. Shaw, the superintendent. As the final heights of lift, toward the end of the work, were greater than had ever been attempted previously, interesting problems were involved. The operating conditions varied between wide limits as the work proceeded; the changing height of lift, ranging from 155 to 1,070 ft., afforded opportunity for experimenting with different percentages of submergence, air pressures, sizes of piping and volumes of water raised per minute. Some tests were made with delivery pipes of uniform diameter; others, with pipes com-

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* P. G. Beckett, *Trans.* American Institute Mining Engineers, Vol. 55, p. 53.

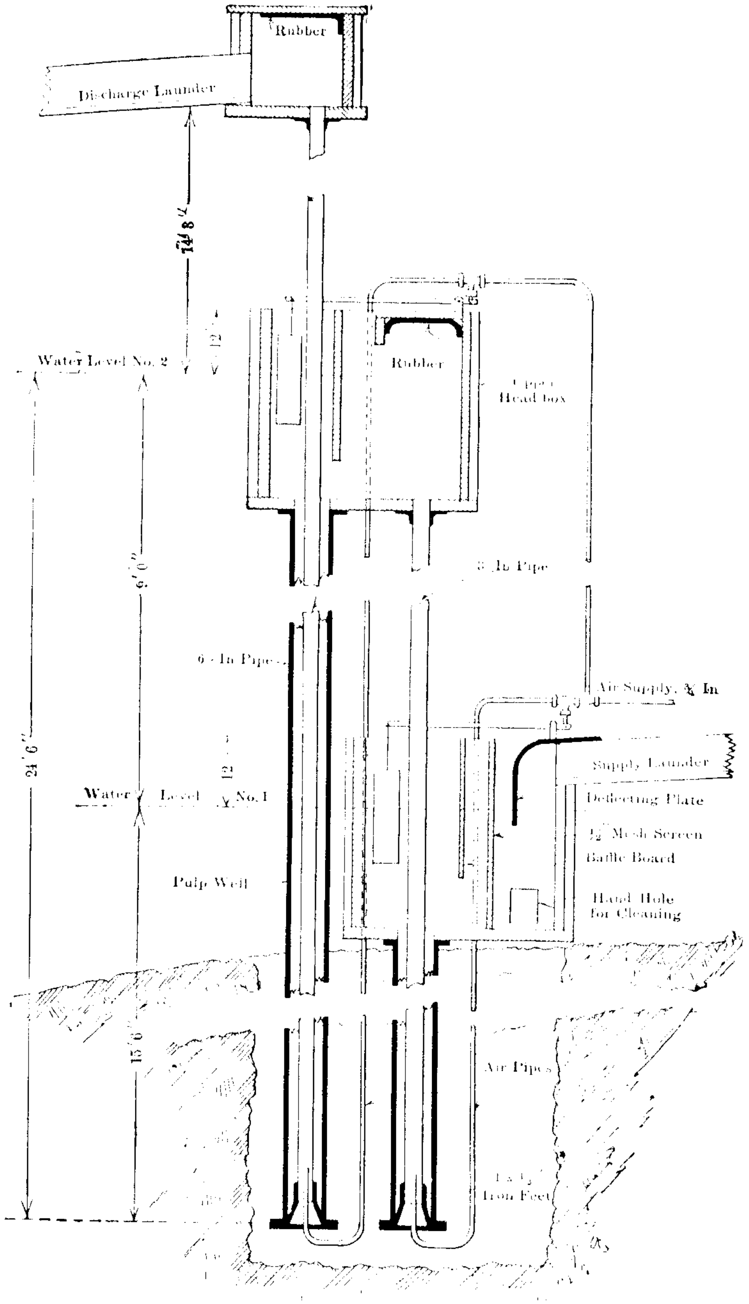

FIG. 10.—Two-stage Air-lift, Coolgardie, Western Australia.

posed of several sections, increasing in diameter toward the discharge end. The expanding delivery pipe usually gave the best efficiency.

Among the outstanding results, it was proved that reasonably good work could be done with much smaller percentages of submergence than had previously been thought practicable. Obviously, however, the greater the submergence the higher the air pressure that can be employed at the foot of the delivery pipe, and the greater the lifting efficiency. In general, in well designed installations: with large submergence (say 30 60 per cent) the efficiency percentage is equal to, or somewhat greater than, the percentage of submergence; but, with very small submergence (0.5 5 per cent) the efficiency percentage is from two to five times the percentage of submergence.

As the details of the large number of experiments and working tests made with these interestng air-lift installations are fully described in a paper by Mr. Shaw (*Trans.* Am. Inst. Min. Engs., Vol. LXIII, pp. 121 456), only a few representative results are presented in Table IX.

TABLE IX.—TESTS ON AIR-LIFTS AT TIRO GENERAL MINES

Lift, feet	Air pressure, pounds	Gallons raised per minute	Cubic feet free air per gallon	Submergence, per cent	Efficiency, per cent
537	98	340	2.90	28.4	37.0
613	108	286	3.30	27.1	35.5
853	102	415	4.38	21.7	37.2
941	71	292	7.42	13.7	29.8
949	61	246	7.76	12.9	....
1,060	....	110	22.00	2.8	30.7
1,070	....	67	29.68	1.8	....

The delivery pipe of the installation from which the above figures were obtained consisted of four sections, 270 ft. each, of 7-, 8-, 9- and 10-in. spiral-riveted pipe. The high air consumption required for the last two smallest submergences should be noted in the light of preceding statements.

Air-lifts for Oil Wells.—Probably the first of these were installed in 1899, at Baku, Russia. Later, the method was introduced into the United States, but did not become important until about 1923.* Air-lifts for raising oil have been most extensively used in the United States in the mid-continent fields (especially Oklahoma and Texas); also in California, Louisana and elsewhere.

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* Previous to this, in some of the older oil fields (Ohio and others), air pressure was applied in an entirely different way, by piping compressed air to the oil-bearing strata. The pressure thus produced in the oil reservoir greatly increased the total percentage recovered from a given area. For data on this method, see: *Scientific American*, March 5, 1921, and *National Petroleum News*, April 5, 1922.

The adoption of this new practice was doubtless stimulated by the results obtained from the deep air-lifts at the Tiro General mine, Mexico (see Table IX). Under the leadership of S. F. Shaw and other petroleum engineers, large numbers of oil wells have been successfully equipped with air-lifts.* The method is practically a *sine qua non* for the numerous recent oil wells that are too deep to permit the employment of plunger or lift pumps operated by rods. There are now many wells from 4,000 to 5,000 ft. deep, and one in California reached a depth, in 1929, of 8,500 ft. The air-lifts in very deep wells must often operate with relatively small percentages of submergence, as shown by the examples given in Table X.

TABLE X.—RESULTS OF TESTS ON AIR-LIFTS IN DEEP OIL WELLS

Locality	Lift, feet	Pipe diameter, inches[1]		Gallons of oil per minute	Cubic feet of air per gallon	Air pressure at foot-piece, pounds	Submergence, per cent	Efficiency, per cent
		Delivery	Air					
Oklahoma.	3,010	6⅝–2	2.0	29.2	29.4	113.0	10.0	14.6
Oklahoma.	4,040	6⅝	2.5	114.0	21.5	250.0	17.4	17.3
Oklahoma.	4,396	6⅝	2.5	0.67	717.0	14.0	0.9	3.5
Peru.	1,840	4.0	8–4.5	47.5	17.5	77.5	12.5	16.2
Colombia.	2,420	4.5	11–4.5	15.5	50.5	80.0	8.5	8.6
Rumania.	4,231	5.0	8–5	6.27	93.8	86.0	5.5	7.8
Rumania.	4,497	3.6	10–3.6	10.8	48.0	158.0	9.8	11.1

[1] In the first three wells, the oil rises through the annular space between the inner tube and casing; in the others, the oil rises through the inner tube.

The natural gas pressure in the oil-bearing strata, which often for a time produces a "flowing" well, can if conserved be utilized to act in connection with the air-lift, the term "air-gas-lift" being then used.

Modes of arranging the piping are: (1) the discharge or flow pipe and the air pipe are separate lines, side by side in the well and connected at the foot-piece; (2) the air pipe is inside of the casing, and the oil rises in the annular space between the two; (3) the air passes down the annular space between the casing and inner tube, the oil rising in the latter. This is the reverse of (2). The first method is most efficient, but there is rarely space enough in an oil well to employ it.

Figs. 5 and 6 of this chapter show examples of different types of foot-pieces for wells of large diameter. In the case of oil wells, the diameter at the bottom is usually so small that a specially designed foot-piece is necessary. Although it is possible to obtain much higher efficiencies than those shown in the tabulated examples (say 40 per cent or higher), oil-well casings are rarely large enough to make high efficiencies practicable.

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* For details of design of piping, foot-pieces and other features, see Bureau of Mines, *Bulletins* 148, 194 and 195, also *Technical Papers* 79, 130 and 235; *Mining and Metallurgy*, Jan., 1926; volume on "Petroleum Technology," published by the Am. Inst. Min. Engrs. in Oct., 1927; also, numerous papers by S. F. Shaw and others published in the Lefax Filing Index, from 1924 to 1928.

CHAPTER XXIV

COMPRESSED-AIR HAULAGE

FOR underground transport, compressed-air and electric locomotives divide between them a broad field of operation. Both are applicable to mine service of all kinds, for hauls of a few hundred feet to several miles. For coal mines containing fire-damp, compressed air is perfectly safe, but electric locomotives must be adopted with caution. Although, by the improvements of recent years, much has been done to prevent serious sparking at the motor, some risk still exists; and, furthermore, the possibility of strong sparking, accompanied by the momentary development of intense heat, from short-circuiting or a ruptured conductor, can hardly be averted.

Besides its safety for gassy, or dry, dusty collieries, or in dry and heavily timbered metal mines, compressed-air haulage has the following advantages: (1) Since the power is stored in the locomotive itself, the system has the maximum degree of flexibility; the locomotives can go wherever track is laid, far beyond the end of the supply-pipe line. Electric locomotives, except those having storage batteries, are dependent upon wiring, which must accompany every foot of advance.* For collieries compressed air may be used equally well for main-line haulage, and for gathering and distributing cars among the working places. (2) Compressed air costs little or nothing when not in actual use, and its full power or but a fraction of it is available at all times. In the intervals between hauls, no power is wasted, because, though the compressor may continue running, it is engaged in storing up power in the pipe-line. A minor consideration is that the exhaust of the locomotive discharges fresh air into the workings, improving the ventilation of the mine.

Compressed-air haulage has certain disadvantages, among which are: (1) the danger of high-pressure air, as in case of breakage of pipe couplings at charging stations; (2) occasional difficulty in starting, when the pressure in the auxiliary reservoir has fallen much below normal, and in controlling starting speed when the main tank pressure is momentarily required.

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* "Cable-reel" or "gathering" electric locomotives are useful for very short distances only.

At most mines compressed air is employed only for underground transport, from the stopes or breasts to the hoisting shaft. In other cases, where the mine is worked through a tunnel or adit, trains are hauled direct to the breaker, tipple, or ore-bins, on the surface. Occasionally, as at the Homestake Mine, Lead, S. D., compressed-air locomotives are used for surface transport of ore, from the crusher houses at the shaft mouths to the different stamp mills, the object being chiefly to reduce the fire risk for the wooden structures, into and near which the haulage tracks pass. For the same reasons compressed-air haulage may be installed at lumber yards, or factories containing inflammable buildings or materials.

Compressed-air locomotives were probably first used by the Plymouth Cordage Co., Plymouth, Mass., about 1873, and in Great Britain, for mine haulage, in 1878. In the United States perhaps twenty compressed-air locomotives were built previous to 1898.* No material changes in design have been made in recent years.

In general terms, a compressed-air haulage plant comprises a three-stage compressor, receiver, pipe-line, charging stations, and one or more locomotives. The locomotive storage tanks are charged with enough air for a round-trip of the maximum length required, after which the locomotive returns to the nearest charging station for a fresh supply. From the main tank the high-pressure air passes through a reducing valve to an auxiliary reservoir, in which the pressure is about 250 lb., as suitable for the initial pressure in the locomotive cylinders.

The Locomotive has one or two storage tanks, which, with the cylinders and other appurtenances, are mounted on a frame carried by four or six driving wheels. Previous to about 1908, practically all locomotives were single-stage. Due to their greater efficiency and economy in the use of the air, compound (two-stage) locomotives are now the rule. Fig. 1 shows a compact design of a four-wheel, compound locomotive, made in four sizes (Table I).

Small compound locomotives, like that in Fig. 1, have the high-pressure cylinder on one side, the low-pressure on the other. Larger sizes (Figs. 3 and 4) have four cylinders, a high- and a low-pressure, tandem, on each side. For ordinary conditions the cylinder volume ratio is 1 : 4. The initial air pressure is usually 250 lb., which, by expansion in the high-pressure cylinder, is reduced to say 50 lb. As the

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* The first compressed-air mine locomotive in the United States was built in 1882 by the Baldwin Locomotive Works, for the Old Eagle colliery, on the Monongahela River, 27 miles above Pittsburgh.

TABLE I.—H. K. PORTER CO.'S TWO-STAGE COMPRESSED-AIR LOCOMOTIVES, CLASS B-P-O

Cylinders: Diameter (in.), high pressure	4½	5½	6	7
Cylinders: Diameter (in.), low pressure	9	11	12	14
Cylinders: Stroke (in.)	10	10	12	14
Diameter of driving wheels (in.)	22	23	24	26
Rigid wheel base (ft. and in.)	2-9	2-9	3-0 or 4-0	4-0
Length over bumpers (ft. and in.)	9 to 12-6	10 to 12-6	12 to 15-6	12 to 18-6
* Extreme width outside gage at cylinders (in.) H.P.	11	12¾	13	14
* Extreme width outside gage at cylinders (in.) L.P.	12¼	14¼	15	16½
Height (see note below)				
Main reservoir capacity (cu.ft.)	20 to 60	30 to 75	60 to 104	60 to 120
Main reservoir length (ft.)	6 to 9	7 to 11	9 to 12-6	9 to 15
Main reservoir diameter (in.)	24 to 36	36 to 40	36 to 40	36 to 40
Main reservoir charging pressure (lb. per sq.in.)	700 to 1,200	700 to 1,200	700 to 1,200	700 to 1,200
Auxiliary reservoir pressure (lb. per sq.in.)	250	250	250	250
Weight in working order (lb.)	7,000 to 10,000	10,000 to 13,000	14,000 to 17,000	18,000 to 22,000
Tractive force (lb.)	1,450	2,200	3,000	4,400
† Hauling capacity—in tons of 2,000 lb. (exclusive of locomotive ; 20 lb. per ton rolling friction):				
On absolute level	68	104	142	210
On 1 per cent grade	32	49	67	100
On 2 per cent grade	19	30	42	63
On 3 per cent grade	13	21	29	45
On 5 per cent grade	7	12	17	26
Weight per yard of lightest rail advised (lb.)	16	16	20	30
Radius of sharpest curve advised (ft.)	15	15	20	30
Radius of sharpest curve practicable (ft.)	12	12	15	20

* Width outside of gage line may be reduced by special construction.

† These hauling capacities are maximum for conditions stated. For satisfactory operation train weights should be 50 to 90 per cent of above figures.

NOTE.—Minimum height is dependent upon gage of track; narrow gages require more height than the wider.

expanded air under these conditions leaves the high-pressure cylinder at about 140° F. below atmospheric temperature (say 75°-80° below zero F.), interheating between the cylinders is always necessary.

Fig. 2 shows diagrammatically an efficient "atmospheric" interheater, consisting of a cylindrical shell filled with small brass or aluminum tubes, between which the exhaust air passes (see also Fig. 1, in which the heater is shown in position). The tubes are open at both ends, the heating medium being atmospheric air, drawn through the tubes by the ejector action of the exhaust from the low-pressure cylinder. As the volume of the high-pressure exhaust air is relatively small, its temperature is thus raised almost to that of the atmosphere.

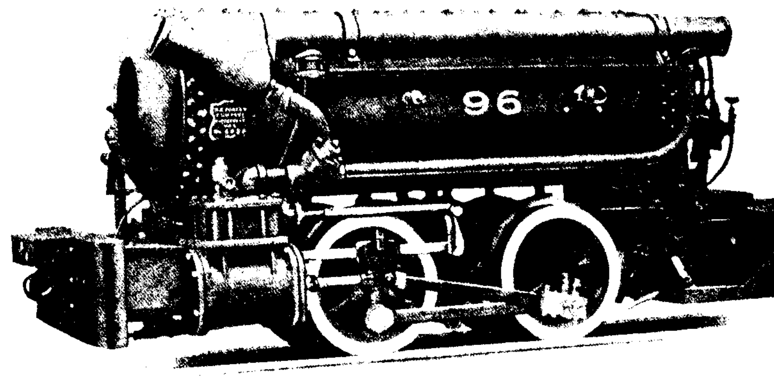

FIG. 1.—H. K. Porter Co.'s Locomotive, Class B-P-O (Table I)

For track with sharp curves, the wheel-base of the locomotive must be short, say 4 ft. 6 in. to 6 ft., for a four-wheel engine. The height of the locomotive over all depends somewhat on the conditions existing in the mine as to thickness of vein, headroom of the haulageways, etc.; it is rarely more than 5 or 6 ft., frequently less. The length varies mainly according to the tank capacity required and the curvature of the gangways. It is usually 10-15 ft. for the smaller sizes, to 20 or 24 ft. for the larger. Widths, 3½-6 ft.

Fig. 3 shows a four-wheel, compound locomotive, made in four sizes (Table II), and Fig. 4 a six-wheel compound (Table III). The hauling capacities given in the tables are maximum for the stated conditions; for satisfactory work, train weights should be 50-90 per cent of those tabulated.

Table II.—H. K. Porter Co.'s Two-stage Compressed-air Locomotives, Class B-PP-O

Cylinders: Diameter (in.), high pressure	7	8½	9½	10
Cylinders: Diameter (in.), low pressure	14	12 and 12	14 and 14	14 and 14
Cylinders: Stroke (in.)	14	14	14	14
Diameter of driving wheels (in.)	26	26	26	26
Rigid wheel-base (ft. and in.)	4–0	4–6	4–6	5–0
Length over bumpers (ft.)	14 to 20	16 to 20	18 to 21	20 to 23
* Extreme width outside gage at cylinders (in.) H.P.	14	15½	16⅜	17¾
* Extreme width outside gage at cylinders (in.) L.P.	16½	16¾	17⅞	17⅞
Extreme width across main reservoir tanks (ft. and in.)	5–5 to 6–5	5–5 to 6–5	5–10 to 6–10	5–10 to 6–10
* Height above rail (ft. and in.)	5–0 to 5–10	5–0 to 5–10	5–0 to 5–10	5–5 to 5–10
Main reservoir, 2 tanks, capacity (cu.ft.)	97 to 160	114 to 210	160 to 275	160 to 290
Main reservoir lengths (ft.)	10 to 18	12 to 18	14 to 20	16 to 21
Main reservoir diameters (in.)	30 to 36	30 to 36	32½ to 38½	32½ to 38½
Main reservoir charging pressure (lb. per sq.in.)	700 to 1,200	700 to 1,200	700 to 1,200	700 to 1,200
Auxiliary reservoir pressure (lb. per sq.in.)	250	250	250	250
Weight in working order (lb.)	20,000 to 27,000	28,000 to 36,000	32,000 to 42,000	43,000 to 46,000
Tractive force (lb.)	4,400	6,400	8,000	9,000
Hauling capacity—in tons of 2,000 lb. (exclusive of locomotive), 20 lb. per ton rolling friction:				
On absolute level	206	305	380	437
On 1 per cent grade	96	145	180	207
On 2 per cent grade	59	92	113	130
On 3 per cent grade	31	65	80	92
On 5 per cent grade	22	38	46	53
Weight per yard of lightest rail advised (lb.)	30	40	45	50
Radius of sharpest curve advised (ft.)	30	35	40	50
Radius of sharpest curve practicable (ft.)	16	18	20	25

* This may be decreased in special cases

Table III.—H. K. Porter Co.'s Two-stage Compressed-air Locomotives, Class C-PP

Cylinders: Diameter (in.) H.P.	7	8½	9½	10
Cylinders: Diameter (in.), L.P.	14	12 and 12	14 and 14	14 and 14
Cylinders: Stroke (in.)	14	14	14	14
Diameter of driving wheels (in.)	26	26	26	26
Rigid wheel-base (ft. and in.)	5–6	5–6	5–6	6–6
Length over bumpers (ft.)	14 to 20	16 to 20	18 to 21	20 to 23
* Extreme width outside gage at cylinders (in.) H.P.	14	15½	16⅜	17⅛
* Extreme width outside gage at cylinders (in.) L.P.	16½	16⅞	17⅞	17⅞
Extreme width across main reservoir tanks (ft. and in.)	5–5 to 6–5	5–5 to 6–5	5–10 to 6–10	5–10 to 6–10
Height above rail (ft. and in.)	5–0 to 5–10	5–0 to 5–10	5–0 to 5–10	5–0 to 5–10
Main reservoir (2 tanks) capacity (cu.ft.)	97 to 160	114 to 210	160 to 275	160 to 290
Main reservoir lengths (ft.)	10 to 18	12 to 18	14 to 20	16 to 21
Main reservoir diameters (in.)	30 to 36	30 to 36	32½ to 38½	32½ to 38½
Main reservoir charging pressure (lb. per sq.in.)	700 to 1,200	700 to 1,200	700 to 1,200	700 to 1,200
Auxiliary reservoir pressure (lb. per sq.in.)	250	250	250	250
Weight in working order (lb.)	20,000 to 27,000	28,000 to 36,000	32,000 to 42,000	43,000 to 46,000
Tractive force (lb.)	4,400	6,400	8,000	9,200
Hauling capacity	The figures in Table II apply here also.			
Weight per yard of lightest rail advised (lb.)	25	30	35	40
Radius of sharpest curve advised (ft.)	50	50	50	70
Radius of sharpest curve practicable (ft.)	30	30	30	50

* This may be decreased in special cases

Fig. 5 shows a small locomotive with 4 by 7-in. cylinders. The operator's seat is detachable, so that the locomotive can be transferred from one level of a mine to another, on a cage with a 5-ft. platform. This design is suitable for small metal mines, or for gathering cars from working places in collieries, to make up trains on main haulageways. For mines having very steep local grades, the H. K. Porter Co. has built locomotives like that in Fig. 1, but with a cable-reel on the rear. This operates like an ordinary locomotive on easy grades, and serves also as a hoisting engine to gather cars from steep cross entries or breasts. The reel, holding 300–1,000 ft. of wire rope, is driven by a pair of small auxiliary cylinders. Fig. 6 is a diagram of the rear end of a locomotive like that in Fig. 1. It shows the course of the air from the main tank

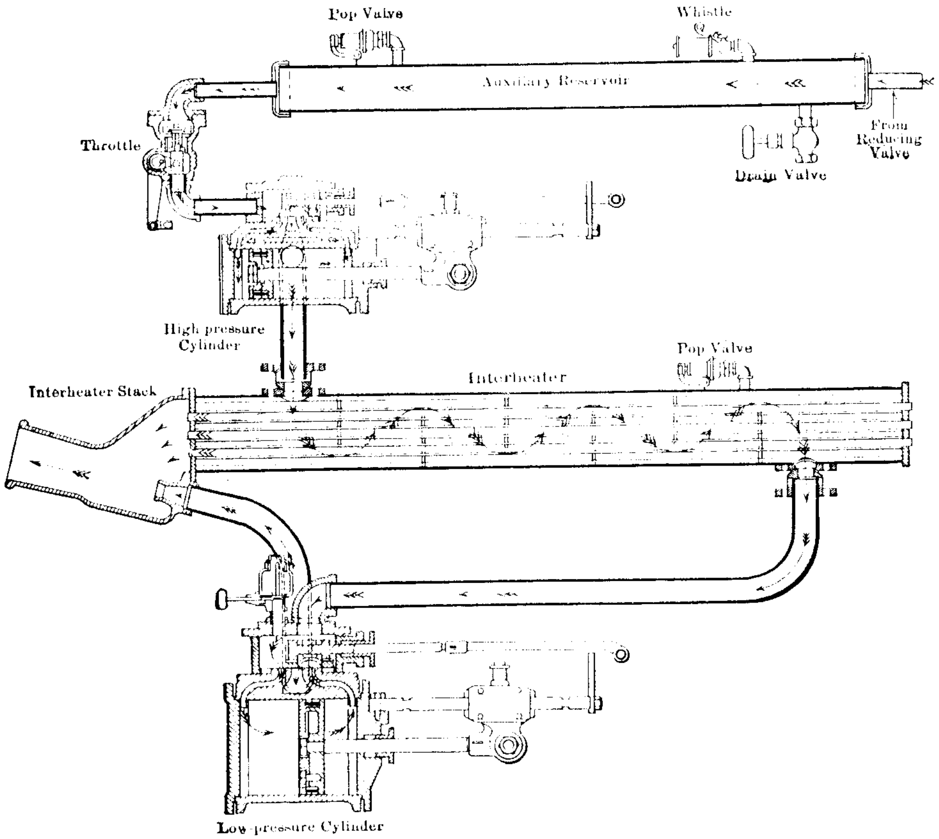

Fig. 2.—Diagram of Interheater and Cylinders for H. K. Porter Co.'s Two-stage Locomotives

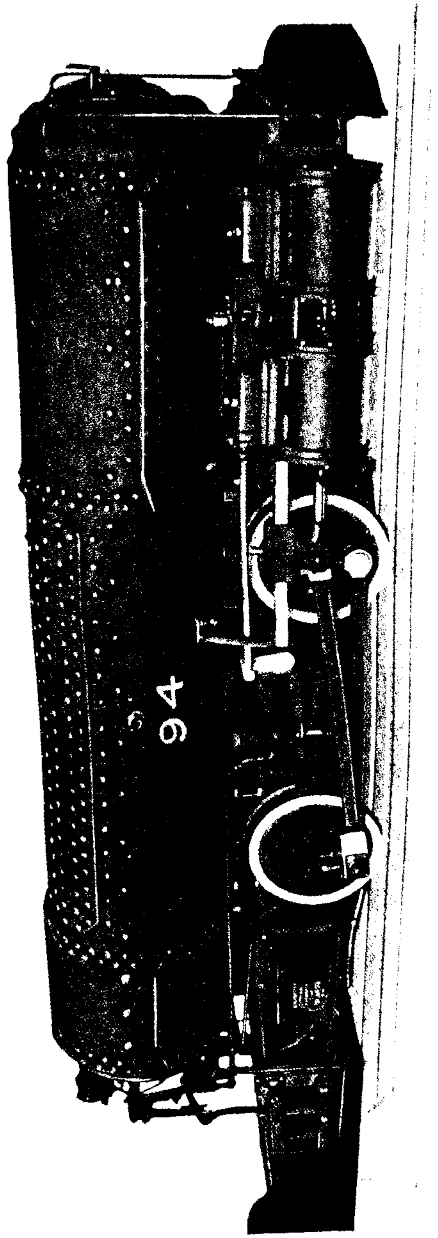

Fig. 3.—H. K. Porter Co.'s Two-stage Locomotive, Class B-PP-O.

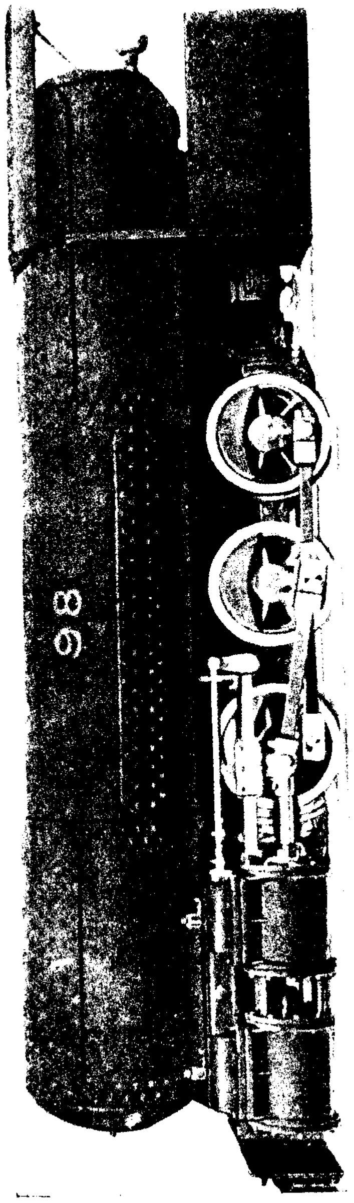

Fig. 4.—H. K. Porter Co.'s Two-stage Locomotive, Class C-PP.

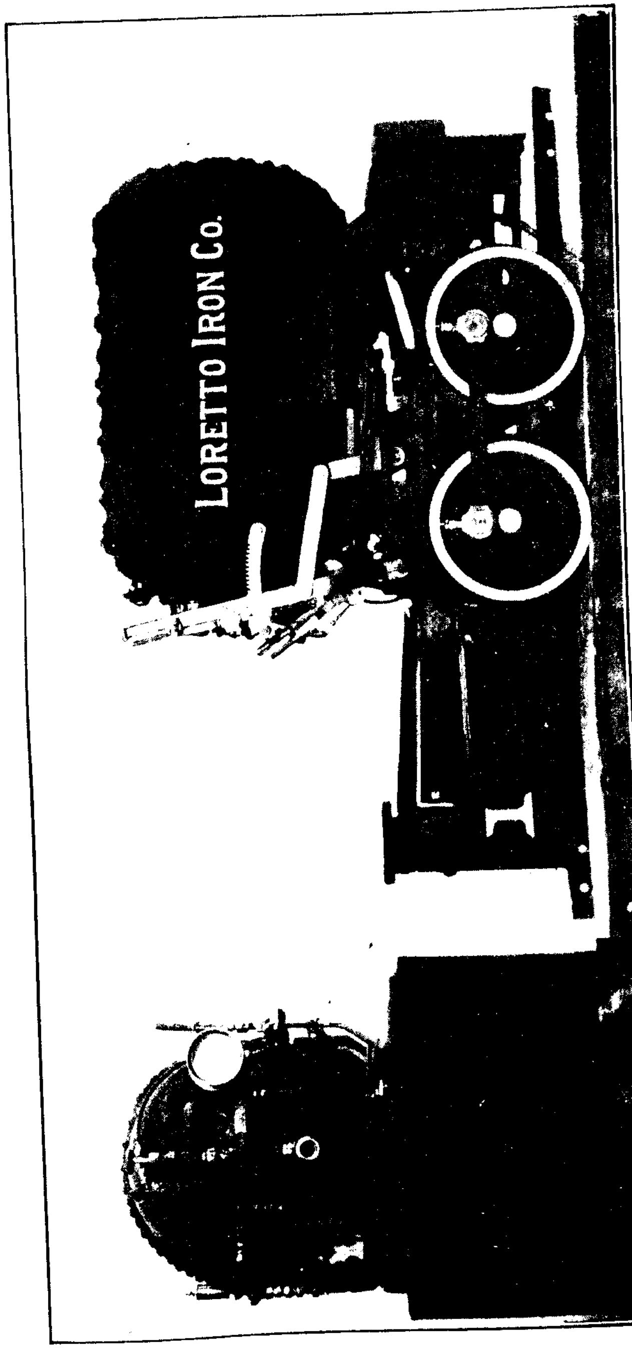

Fig. 5.—H. K. Porter Co.'s Locomotive for Small-scale Work.

to the auxiliary reservoir, thence through a by-pass valve to the reheater, and finally to the cylinders.

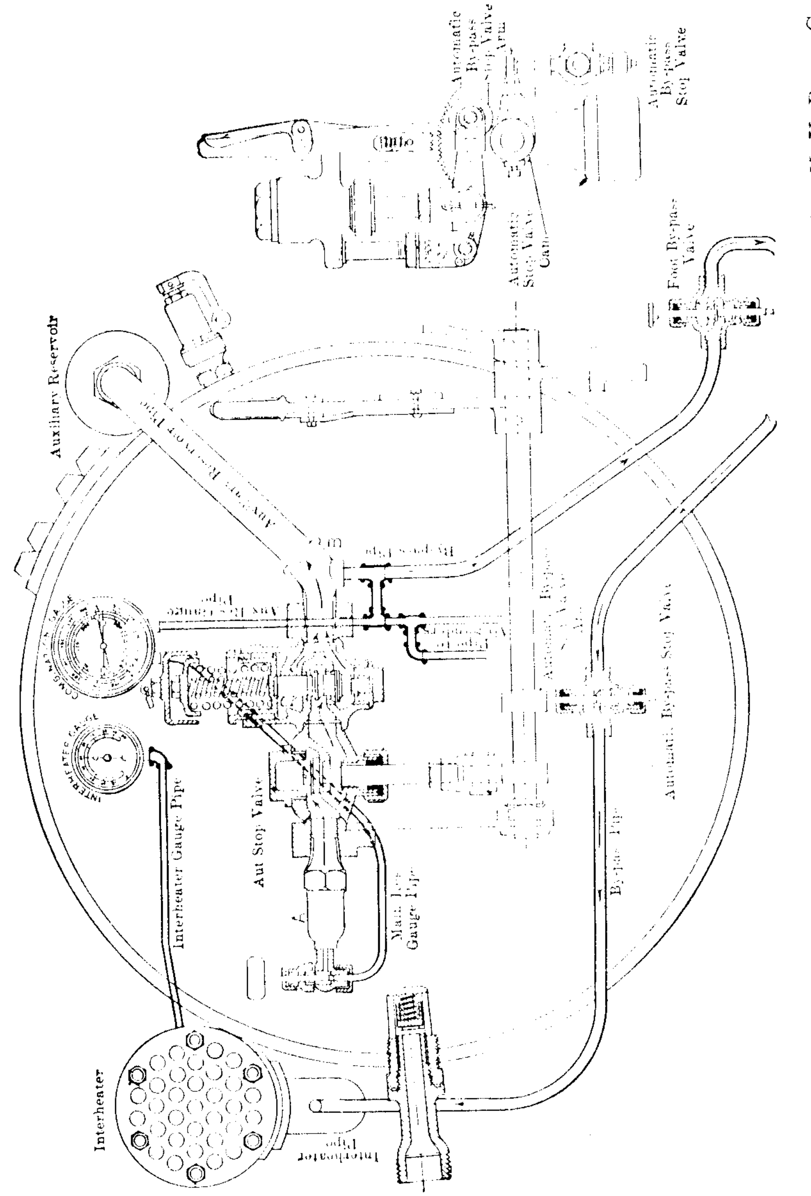

Fig. 6.—Diagram of Air Piping, Interheater, Valves, etc., on Rear End of Two-stage Locomotive (H. K. Porter Co.)

Locomotive Details.—The tanks have dished or approximately hemispherical ends, and are built of heavy boiler plate, with a tensile strength of 60,000–68,000 lb. per sq. in.; the shells are ¾ to ⅞ in. thick, with 1 to 1¼-in. heads. Ring seams are double riveted with lap joints; longitudinal seams have butt joints, with inside and outside welt strips. As the tanks are generally designed for working pressures of 700–900 lb., the longitudinal seams have 6–8 rows of rivets, to make a joint of not less than 75 per cent of the strength of the plate. Tanks are tested with a pressure about 30 per cent greater than the working pressure, the factor of safety with plate of the usual quality being about 3½; this is sufficient, as there are no expansion and contraction strains as in a boiler. For very high pressures, tanks of large diameter are unsafe, and are replaced by several smaller tanks (Fig. 7), or by a set of heavy seamless steel tubes, 8 to 9 in. in diameter. Tubes 9 in. in diameter by $\frac{11}{32}$ in. thick will carry pressures of 2,000–2,200 lb. per sq. in. These high pressures are unnecessary for ordinary haulage service.

From the main tank the air passes into a small auxiliary reservoir and thence to the cylinders. This reservoir is usually a section of wrought-iron pipe, 4 to 9 in. in diameter and 6 to 15 ft. long, laid alongside the main tank. The pressure in it is adjusted by an automatic reducing valve to the requirements of the engine—usually 140 to 150 lb. for single-stage, and 250 to 300 lb. for two-stage locomotives, depending on the size of cylinders and the power required. On the locomotives of the H. K. Porter Co., the reducing valve is a double-seated balanced valve, operated by a small piston. The air pressure in the reservoir acts on one side of the piston, tending to close the valve. This action is opposed by an external spring, adjusted to keep the valve open until normal working pressure is reached in the reservoir. Then the valve is closed by the air pressure, against the spring resistance. To provide for the case when the locomotive is using no air (as on a down grade or when at rest), a single-seated supplementary valve is placed in the pipe between the reducing valve and the main tank. This valve is controlled by the throttle lever; being open when the throttle is open, otherwise closed by the air pressure. Having the two valves, leakage

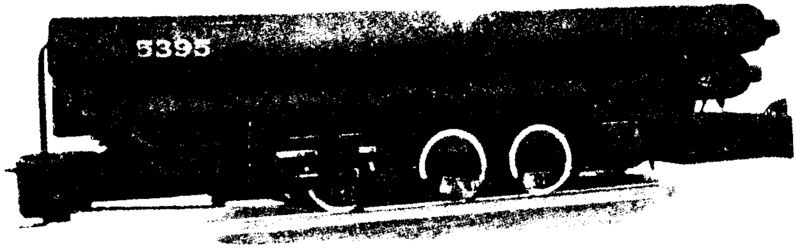

Fig. 7.—H. K. Porter Co.'s Locomotive, Class C–5Ps–O.

from main tanks to auxiliary reservoir is avoided and a close regulation secured.

From the auxiliary reservoir the air passes to the cylinders through a balanced throttle valve. This maintains a constant working pressure, suited to the needs of the locomotive, prevents waste of air likely to ensue if air at full tank pressure were admitted to the cylinders, and makes the locomotive more manageable. In starting a heavy load excessive slipping of the drivers is avoided, and with light loads the reducing valve may be regulated for any desired pressure. Toward the end of the trip, when the pressure in the main tanks falls to that in the auxiliary, the cylinders take air directly from the former, and the locomotive will continue to run as long as the pressure remains sufficient. For long hauls, or when the cross-sectional dimensions or sharp curves, or both, of the haulage-ways do not permit the use of tanks of great length or large diameter, a tender carrying a supplementary tank may be employed (Fig. 8).

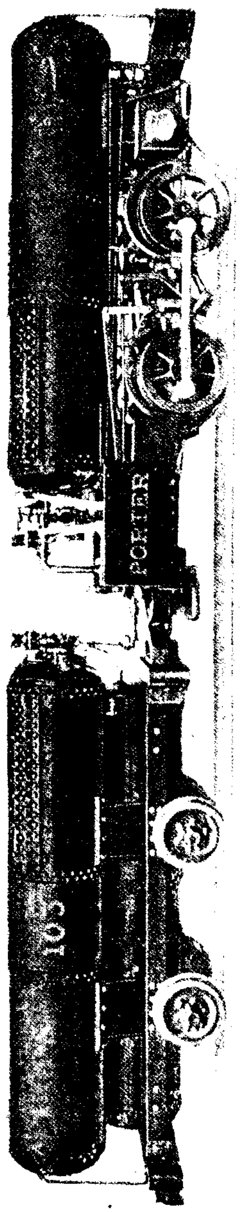

FIG. 8.—H. K. Porter Co.'s Locomotive, Class B-PP-T (Cylinders are inside of the frame).

For small, single-stage locomotives, the air is sometimes admitted to the cylinders throughout nearly full stroke, and consequently, as the exhaust is at high pressure, the efficiency is low. This practice is due to the tendency to use as small a locomotive as possible, on account of the limited headroom and narrow, crooked gangways so common in mines. Better results are obtained by working with a cutoff, and increasing the size of the cylinders and the weight on the drivers. In using air expansively, as can be done with properly proportioned cylinders, there should be no trouble from freezing of moisture. Although expansion produces a low cylinder temperature, yet, as the initial pressure is so much higher than is employed for other compressed-air machinery, the expanded air is relatively dry, and the force of the exhaust is sufficient to keep the ports clear of ice. To this end the ports should be large, straight, and short. The cylinders are not lagged with non-conducting covering, as is so necessary for steam cylinders, to minimize condensation. By exposing their surface to the warm air of the mine, some heat is absorbed. Occasionally the exterior surface of the cylinders is cast with deep corrugations, to present a large area to the warm surrounding air. The cylinders are provided with slide valves; piston valves, like those used in steam locomotives, would leak more because of the dryness of the air.

On account of the cold produced by reducing the pressure between the main tank and auxiliary reservoir, and to increase efficiency of operation, reheating is advantageous (though not essential) for single-stage locomotives. It is best done by heating the auxiliary reservoir. If steam is available in the mine, a little steam and hot water may be injected into the reservoir each time the locomotive is charged. Or, in mines free from fire-damp, a small reheater, burning oil or coke, may be carried on the locomotive. When the air is reheated water should always be kept in the small tank; the moisture from it, passing with the air into the cylinders, assists in lubricating the valves and pistons. For compound locomotives interheating only is necessary, applied between the cylinders (see Fig. 2 and accompanying description).

Pipe-line and Charging Stations.—The required capacity of the plant depends on the length of haul and size of locomotives, as influenced by the daily output, weight of trains, and track gradients. For short hauls, the pipe-line is sometimes omitted, the locomotive returning to the compressor receiver to be recharged. Usually a pipe-line is carried underground, with charging stations at one or more points, located according to the haulage distances and the capacity of the locomotive tanks. The innermost station, farthest from the compressor, must be at a point from which the locomotive can reach the end of its trip and return for recharging. For long hauls, heavy traffic, or adverse gradients, several charging stations may be required. Inside receivers are unnecessary, unless the diameter of the pipe-line is too small. The pipe-line itself acts as a storage reservoir, and should be of a diameter which, in proportion to its length, will furnish a cubic capacity sufficient to charge the locomotive tank quickly and without excessive drop in pressure. That is, the pressure in the tank and pipe-line on equalizing should not fall much below the pressure which the locomotive is designed to carry. To this end, the volume of pipe-line storage should be at least three times the tank capacity of the locomotive. To determine the necessary pipe-line capacity, several variables must be harmonized, as follows:*

V = storage volume required, cubic feet;
v = volume of locomotive tank, cubic feet;
P = pipe-line pressure, pounds per square inch (usually 900–1,200 lb.);
p = desired pressure in locomotive tank, pounds per square inch (700–900 lb.);
p' = residual pressure in locomotive tank, just before recharging, pounds per square inch.

Then: $V(P - p) = v(p - p')$, or $V = \frac{v(p - p')}{P - p}$.

For example, let P = 900 lb., p = 750 lb., p' = 125 lb., and v = 100 cu.ft., from which:

$$V = \frac{100(750 - 125)}{900 - 750} = 416.6 \text{ cu.ft.}$$

By transposition, the same formula serves for finding the pipe-line pressure required for a given tank pressure. When there are several locomotives, it is rarely necessary to design the pipe-line for charging more than one at a time. If the volumetric capacity of the pipe-line is ample, the drop in gage pressure on charging is soon recovered by the compressor, which, except in plants operating a single locomotive, is kept in nearly constant operation. If more locomotives are added after the original installation of the system, the same pipe-line may still serve, provided the compressor is able to charge it to full pressure at shorter intervals

The piping, which is generally from 3 to 5 in., should be of the best material, lap-welded, and should have sleeve joints made with the utmost care to prevent leakage. To stop leaks, the sleeves should have annular grooves (counter-bores) at each end, into which lead calking is driven if required. About every 300 ft. a pair of flanges should be placed in the line, and valves at suitable places, for convenience in making repairs or extensions. There should also be a valve between the compressor and pipe-line, so that the compressor can be inspected or repaired without wasting the air in the pipe. The pipe should not be buried, but carried along one side of the haulageway, either on the floor or on brackets, so that leaks will at once attract attention. An occasional bend in the pipe-line is advantageous in permitting free expansion and contraction, but bends should not be too numerous, as they involve

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* H. K. Porter Co., "Modern Compressed-Air Locomotives."

more joints and greater possibility of leakage. Pipe-lines should be painted with some non-corrosive coating.

Charging Stations.—Fig. 9 shows one design, consisting of a heavy tee inserted in the air main, with a flexible connection for coupling to the locomotive. The connecting pipe comprises a vertical, rigid branch, with a $1\frac{1}{2}$-in. gate-valve, and a short horizontal pipe, with a union and a ball-and-socket joint. A further flexible connection, with two ball joints, serves for coupling to the locomotive tank. Thus, for charging, the locomotive need not be spotted accurately on its track, but has a foot or two leeway. When not in use, the flexible connection is swung aside.

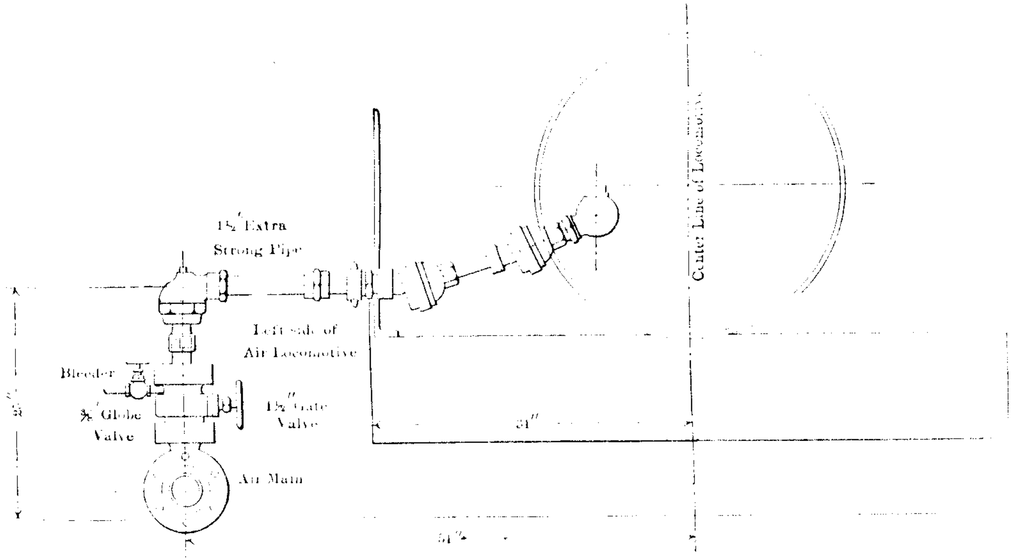
$1\frac{1}{2}''$ Extra Strong Pipe
Left side of Air Locomotive
Bleeder
$\frac{3}{8}''$ Globe Valve
$1\frac{1}{2}''$ Gate Valve
Air Main
Center Line of Locomotive
31"

Fig. 9. Compressed-air Locomotive Charging Station.

After coupling to the locomotive, the gate-valve is opened, whereupon the air pressure immediately forces together the parts of the ball-and-socket joints and makes a tight connection. When equilibrium is established between the pressures in the pipe-line and tank, the gate-valve is closed. To break the coupling, the compressed air remaining in the connecting pipe, between the gate-valve and the locomotive check-valve, must first be released. This is done by opening a small "bleeder valve," placed just above the gate-valve. The joints then loosen and are readily manipulated. The actual time occupied in charging is usually about three-quarters of a minute, but, including stopping the locomotive and making the connection, $1\frac{1}{2}$-2 minutes may be allowed. Charging may often be done while shifting cars and making up trains. Fig. 10 shows another form of charging station.

Calculation of Motive Power.—Several factors must be known: the tractive resistance per ton of the loaded cars on a level, resistances due

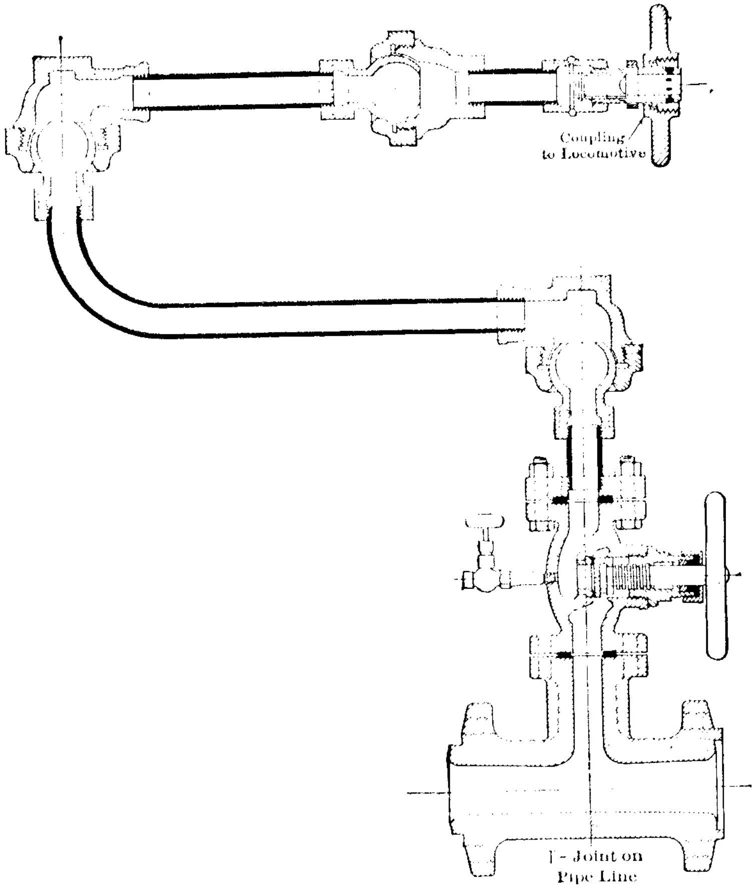

Fig. 10.—Charging Station (H. K. Porter Co.)

to gradients and curves, weight of empty and of loaded cars, and number of cars to be hauled per train. The values of these factors are readily ascertained, with exception of the resistances due to curvature of track and character of roadbed. The former has been determined experimentally for ordinary surface railways, but mine track is apt to be roughly laid, with curves of varying and irregular radius, and the elevation of the outer rail improperly adjusted.

The average tractive force required per ton depends on the condition of the track and roadbed, and the character and state of repair of the running gear of the cars. On level mine track the coefficient of rolling friction is usually from 30 to 40 lb. per ton, though it may be considerably higher on poorly laid or light track. With mine track in exceptionally good condition, this coefficient may be as low as 20 lb. The grade resistance is 20 lb. per short ton, for each 1 per cent of grade. The distribution of grades is often such that the maximum load is not the resistance of the loaded trains, which are usually hauled on slight down grades, but that of the return trains of empty cars on adverse gradients. For the most economical work, gradients should not exceed $\frac{1}{2}$ to $\frac{3}{4}$ per cent in favor of the loaded trains. With ordinary track and rolling stock, and a grade of 5 or 6 in. per 100 ft., the coefficient of rolling friction is nearly the same for a loaded train hauled down as for an empty train of the same number of cars hauled up the grade. Heavier adverse grades are often necessary ($2\frac{1}{2}$ or 3 per cent or more), but they should be avoided, because the maximum tractive force of a locomotive falls off rapidly. On a $2\frac{1}{2}$ per cent adverse grade the locomotive can haul only about four times its own weight, even if the track is not slippery. Grades should be reduced on curves. Colliery cars carrying $2\frac{1}{2}$ to $3\frac{1}{2}$ tons weigh 1,800 to 2,300 lb., whereas those used in metalliferous mines, for mechanical haulage, weigh 1,000 and 2,000 lb. Having ascertained the values of the different factors, the proper allowance of reserve power, in terms of volume and pressure of air, to cover indeterminate resistances due to local imperfections of track and rolling stock, is a matter of judgment and experience.

With a given air pressure, the capacity of the locomotive tanks depends primarily on the length of round trip to be made with a single charge of air. When this distance is 1 or $1\frac{1}{2}$ miles, the tank capacity is generally from 50 to 150 cu.ft., according to the load; which, in turn, together with the track and grade resistances, governs the cylinder dimensions. Cylinders of 5 by 10 in. up to 9 by 14 in. are common, the larger sizes being for heavy work in collieries or on surface. For runs exceeding $1\frac{1}{2}$ miles, it is often desirable to increase the air pressure, rather than the tank capacity. Another plan is to provide a tender, carrying an auxiliary tank (Fig. 8).

Having determined the total foot-pounds of work to be done with a single charge of air, on a run of the maximum length, specifications may be obtained from the builders for a locomotive of suitable weight, gage, wheel-base, tank capacity, and cylinder dimensions.

Compressors for Charging Locomotives are usually three-stage, occasionally four-stage. The air cylinders of the higher stages are single-acting. Examples are shown by Figs. 30 and 31, Chapter III, with their accompanying specifications.

When the mine is already provided with a low-pressure air plant, for machine drills, etc., and if it has some surplus capacity, a two-stage charging compressor may be installed, to take air from the low-pressure system and bring it up to the tension required for the locomotives. Care must be exercised in making such a combination, and it is not advisable. If the quantity of air from the low-pressure compressor should at times be insufficient to furnish the necessary excess for the locomotive charger, the latter might have to compress from too low an initial pressure. There would then be excessive development of heat which might raise the cylinder temperature to the flashing-point of the oil, thus causing an explosion (Chapter XIII).

The capacity of the charging compressor depends on the pipe-line pressure, number of locomotives to be operated, cubic contents of the locomotive tanks, pressure carried by the system, and the relation between charging periods.

Let C = compressor capacity required, cubic feet free air per minute;

L = locomotive-tank capacity, cubic feet free air per minute;

N = number of charges required in any given time, T.

Hence the equation: $C = \dfrac{NL}{T}$.

For example, if $N = 3$, $L = 5{,}200$ (corresponding to 100 cu.ft. of air at 750 lb. gage pressure), and $T = 60$ minutes:

$$C = \frac{3 \times 5{,}200}{60} = 260 \text{ cu.ft. free air per minute.}$$

When the locomotives are charged at approximately equal intervals of time, a single application of the above formula will be sufficient. Otherwise, calculations are required to determine the maximum and minimum rates of consumption of air. For plants installed at an altitude above sea-level, allowance must be made for decreased compressor output (Chapter XII).

Examples of Compressed-air Haulage Plants are given in Table IV.

The following data are on a plant of the Philadelphia & Reading Coal & Iron Co.:

(*a*) Shaft level—1 locomotive.

Round trip 5,200 ft.; grades $\frac{4}{10}$ to $\frac{8}{10}$ per cent, all in favor of load; charging station at each end of run; gage of track, 14 in.; weight of cars, empty, 3,300 lb.,

TABLE IV.—DATA ON COMPRESSED-AIR HAULAGE PLANTS (Dec., 1926

	U. S. S. R. & Mining Co., Bingham Canyon, Utah		Ray Mine, Arizona	Inspiration Mine, Arizona	Philadelphia & Reading Coal & Iron Co.	Homestake Mine, South Dakota			
	Main	Intermediate							
Number of locomotives in use	3*	3†	23§	18	42	2	2	20	10
Weight of locomotive, pounds	16,000	8,000	16,500	20,000	14,000–25,700	28,000¶ 26,000	17,000	10 000	7,000
Dimensions, inches:									
Width	48	45	58	64	79	63	52	40	36
Height	65	58	61	69	64	82	76	61	57
Length	164	127	208	192	192	276 and 256	178	106–135	106
Track gage, inches	18	18 and 24	30	30	44 and 48	22	22	18	18
Draw-bar pull, pounds	3,000	1,450	3,000	5,000	3,500–4,500	6,400	3,000	2,200	1,450
Capacity, main tank, cubic feet	90	40	82	105	78–110	165 and 137	70	28.5–40	20.5
Capacity auxiliary tank, cubic inches			2,488		5,040	4,368 3,456	2,550	990–1,340	640
Tank pressure, pounds	900	900	850	800	900	900	900	900	900
Initial cylinder pressure, pounds	250	250	200	250	150–300	250 and 170	170	145	145
Cars per train	7	8	20–30	25 (max.)	8–18	31	11	25	12
Weight of car, pounds	8,300	1,900	3,600	3,900–5,475	5,000	5,100	5,100	1,200–1,600	
Weight of car contents, pounds	16,000	4,000	10,000	9,000–10,000	8,550	9,400	9,400	2,000	2,000
Cost of haulage per ton-mile, cents	10.56 ‡	13.05 ‡		13.0	4.0				

NOTES.—All locomotives listed above were built by the H. K. Porter Co., except 10 of those in the Philadelphia & Reading Coal & Iron Co.'s mines, which are by the Baldwin Locomotive Works. Locomotives working with initial cylinder pressures of 250 lb. and over have compound cylinders; the others have simple cylinders.

* Cylinders, 6 by 12 by 12 in.

† Cylinders, 4½ by 9 by 10 in.

‡ Total cost except depreciation.

§ Average haul, 0.83 mile (in this mine air haulage is being gradually superseded by electric haulage).

¶ For surface haulage, shafts to ore bins.

loaded, 8,800 lb.; 15-38 cars per trip; total output, 600 cars per 10 hours. Round-trip time, 12 min.; charging time, 1 min. A round trip and a half can be made with one charging.

(*b*) Slope level—1 locomotive.

Length of haul, 3,200 ft., of which 700 ft. is on an adverse grade of $4\frac{1}{10}$ to $5\frac{1}{2}$ per cent. Grade of main gangway, $\frac{4}{10}$ to $\frac{8}{10}$ of 1 per cent, in favor of load. Trains of 10 cars are hauled on main gangway, and 4 cars on the slope; weights of cars same as above.

Locomotive-tank pressure at start, 600 lb.; at end of trip, 200 lb. Average working pressure, 180 lb.

The cost of the plant and its operation was as follows (pre-war prices):

One Norwalk 3-stage compressor, erected	$5,180 74
Pipe-line, 4,200 ft., 5 in., including three charging stations	2,951 06
Two Baldwin compressed-air locomotives and fittings	4,904 33
Alterations in gangways to adapt them to locomotive haulage	665 17
Total cost	$13,701 30
Daily operating cost, for 180 days in the year	$14 69
Fixed charges, depreciation, repairs, etc., figured at 10 per cent, together with cost of steam power	9 00
Total running expenses per day	$23 69
Haulage cost per car, at 660 cars per day	3 6 cents
Previous cost of mule haulage per car	5 1 cents
Saving per year, about	$1,800.00

For full operating details and costs of an old compressed-air haulage plant at a Pennsylvania anthracite mine, see *Trans.* Amer. Inst. Min. Engs., Vol. XXX, p. 566.

CHAPTER XXV

MEASUREMENT OF AIR CONSUMPTION

A DISCUSSION of the transmission of compressed air through long pipes, with formulas and numerical data, is given in Chapter XV. The behavior of air is different when flowing through short pipes, or orifices in thin plates, and under pressures relatively small as compared with those existing in ordinary compressed-air transmission.

Flow of Air through Orifices or Short Tubes. The fundamental equation is $v = c\sqrt{2gh}$, in which: v is the velocity of flow, feet per second; g, the force of gravity, or 32.2; h, the height in feet of a column or head of air, required to produce the pressure under which the air is flowing, that is, $h = p^1 - p^2$ = difference between the pressures on the two sides of the plate containing the orifice; and c, a coefficient of flow, determined experimentally. The volume discharged, in cubic feet per second, is equal to v multiplied by the area of the orifice in square feet.

Values of constant "c" (Weisbach, Mechanics of Engineering, pp. 945-947.)

FLOW THROUGH AN ORIFICE IN A THIN PLATE

Diameter of orifice = 1 cm. = 0.394 in.

Ratio of pressures on the two sides of the orifice = $\frac{p^1}{p^2}$	1.05	1.09	1.43	1.65	1.89	2.15
Values of c	0.555	0.589	0.692	0.724	0.754	0.788

Diameter of orifice = 2.14 cm. = 0.843 in.

Ratio of pressures	1.05	1.09	1.36	1.67	2.01
Values of c	0.558	0.573	0.634	0.678	0.723

FLOW THROUGH A SHORT TUBE

Diameter of tube = 1 cm. = 0.394 in.; length 3 cm. = 1.181 in.

Ratio of pressures = $\frac{p^1}{p^2}$	1.05	1.10	1.30
Values of c	0.730	0.771	0.839

Diameter of tube = 1.414 cm. = 0.557 in.; length, 4.242 cm. = 1.670 in.

Ratio of pressures	1.41	1.69
Values of *c*	0.813	0.822

Diameter of tube = 1 cm. = 0.394 in.; length, 1.6 cm. = 0.630 in. Edge of orifice rounded.

Ratio of pressures	1.24	1.38	1.59	1.85	2.14
Values of *c*	0.979	0.986	0.965	0.971	0.978

In this case *c* always approximates unity. A short conical pipe, rounded off at the inlet orifice, gave nearly the same values for *c*.

An empirical formula for the velocity of flow, suitable for dealing with the small differences of pressure used in ventilation, is (Clark's Rules, Tables and Data, p. 891):

$$V = C\sqrt{\frac{2gh}{12} \times 773.2 \times \left(1 + \frac{t - 32}{493}\right) \times \frac{29.92}{p}}$$

$$= 352C\sqrt{[1 + 0.00203(t - 32)]\frac{h}{p}}$$

where: V = velocity of flow, feet per second; C = coefficient of efflux; $g = 32.2$; h = inches of water column measuring the difference of pressure; t = the temperature, Fahr.; p = barometric pressure, in inches of mercury. The constant 773.2 is the volume of air at 32°, under a pressure of 29.92 inches of mercury, when the volume of an equal weight of water is taken as 1.

For 62° F., $V = 363C\sqrt{\frac{h}{p}}$, and if $p = 29.92$, $V = 66.35C\sqrt{h}$.

In this formula, the values of C (Weisbach), for pressures of 0.23 to 1.1 atmosphere, are:

For circular orifices in thin plates	0.56 to 0.79
For short cylindrical tubes	0.81 to 0.84
For the same, rounded at the inner end	0.92 to 0.93
For conoidal mouthpieces, in form of the "vena contracta"	0.97 to 0.99
For conical converging mouthpieces	0.90 to 0.99

Measurement of Low-pressure Air.—Fig. 1 shows an apparatus for measuring small quantities of air at low pressure.[*] It can be used for any ordinary working pressure, as employed for mine service, but the

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[*] G. S. Weymouth and C. C. Freeman. *Mining and Scientific Press*, April 20, 1912, p. 562.

TABLE I.—ACTUAL DISCHARGE, POUNDS OF DRY AIR PER SECOND, AT 60° F. AND 14.7 LB. BAROMETRIC PRESSURE, FOR CIRCULAR ORIFICES IN A PLATE 0.057 IN. THICK (1 LB. OF AIR UNDER THE ABOVE CONDITIONS = 13.09 CU.FT.)* Kent.

Pressure, inches of water column†	Diameter of orifice, inches								
	0.3125	0.500	1.000	1.500	2.000	2.500	3.000	3.500	4.000
½	0.00114	0.00293	0.0117	0.0263	0.0468	0.073	0.105	0.143	0.187
1	.00162	.00416	.0166	.0373	.0663	.103	.149	.202	.264
1½	.00199	.00510	.0203	.0457	.0811	.127	.182	.248	.323
2	.00231	.00590	.0235	.0538	.0937	.146	.210	.285	.373
2½	.00259	.00662	.0263	.0591	.1050	.163	.235	.319	.416
3	.00285	.00726	.0289	.0648	.1150	.179	.257	.349	.455
3½	.00308	.00786	.0312	.0700	.1240	.193	.277	.377	.491
4	.00330	.00842	.0334	.0749	.1330	.206	.296	.402	.525
4½	.00351	.00895	.0355	.0794	.1410	.219	.314	.426	.556
5	.00371	.00945	.0375	.0838	.1480	.231	.331	.449	.586
5½	.00390	.00993	.0393	.0879	.1550	.242	.347	.471	.613
6	.00408	.01019	.0411	.0918	.1620	.252	.362	.492	.640

* The general expression for the weight of dry air is: $W = \frac{1.325B}{T}$, where W = weight of 1 cu.ft. of air; B = absolute pressure, inches of mercury; T = absolute temperature (F.)

† 1 inch of water column = 0.0361 lb. per sq. in. = 5.20 lb. per sq. ft.

actual measurement is applied to an equal flow of air under a low pressure. That is, for convenience in measuring while carrying on the test, an equal volume of flow of low-pressure air is substituted for the regular high-pressure supply. The apparatus here described is for dealing with small volumes, say 20 cu.ft., or less, of free air per minute; but, by using larger tanks, any quantity can be measured, provided the supply pressure is constant during the test and the flow regular for the few seconds necessary to obtain the first gage reading. The volume of air thus isolated is then measured at leisure. In using this method, the regular supply must be diverted for about 0.5 minute.

Connected with the air supply is a vertical 1-in. pipe, with globe-valves *A* and *D*, stop-cock *B*, and a mercury gage *I*. This pipe leads by a hose to a closed water tank *K*, of about 30 cu.ft. capacity, which is provided with a gage glass and a short open hose *H* from the bottom. A branch pipe, with stop-cock *C* and globe-valve *E*, leads to a 5-gal. drum (of light iron or tin plate) connected with which is a water-gage *J*. The drum has an opening at *F*, to which disks with orifices of different sizes can be applied, the area of orifice being such as to keep the pressure in the drum low enough to eliminate factors caused by variations of volume due to pressure. These orifices may be of, say, $\frac{1}{8}$, $\frac{1}{4}$, $\frac{1}{2}$ and $\frac{3}{4}$ sq.in. areas.

To make a test, valve *A* is opened to admit the necessary quantity of air; cock *C* is then opened and valve *E* regulated to give any required pressure on the mercury gage *I* (for example, 5 lb. per sq.in.). An orifice is attached at *F* to the 5-gal. drum, to give a reading on the water-gage *J* of 2 to 9 in. (= 0.0722 to 0.325 lb. per sq.in.). The hose at *G* is disconnected, cock *B* opened, cock *C* closed, and valve *D* regulated to give the same reading on gage *I* as before; thus, the resistance of valve *D* is made equal to that of valve *E*, leading to the orifice at *F*. Tank *K* is then nearly filled with water, and its level noted on the gage glass; cock *C* is opened, *B* is closed, and the hose is connected at *G* to tank *K*. Next, cock *B* is opened and *C* closed simultaneously, and the free end of hose *H* lowered; the reading on gage *I* is noted and, when the tank is nearly empty, *C* is opened and *B* closed simultaneously. The time the air is passing to tank *K* is taken by stop-watch; the air in *K* is reduced to atmospheric pressure (by bringing the free end of hose *H* to the water level in the tank), and the volume of air noted. If necessary, due to any back pressure from tank *K*, gage *I* is brought to the same level as when air was passing to *K*, by manipulating valve *E*, and a reading on *J* is taken.

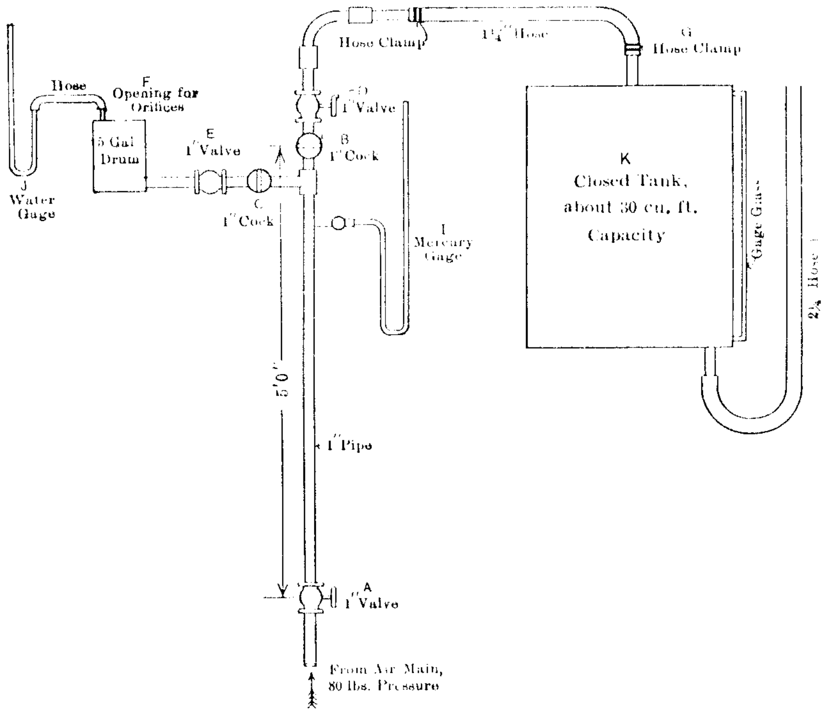

FIG. 1.—Diagram of Air Apparatus for Measuring Low-pressure Air (Weymouth and Freeman).

Table II shows the results of measurements, using orifices of 0.214, 0.441 and 0.797 sq.in., a pressure on the mercury gage equal to 9 lb. per sq.in., and volumes of free air of 9 to 16 cu.ft. per minute. The volumes discharged through the orifices, computed from the formula $v = c\sqrt{2gh}$, check closely with those measured in the tank. In using the formula, the coefficient of discharge was taken as 0.64 and the weight of 1 cu.ft. of air at 62° F., as 0.0671 lb.

TABLE II.—RESULTS OF MEASUREMENTS OF LOW-PRESSURE AIR BY APPARATUS OF G. S. WEYMOUTH AND C. C. FREEMAN. (*Mining and Scientific Press*, April 20, 1912.)

Orifice F, square inch	Mercury gage *I*, inches	Water gage *J*, inches	Calculated Volume, Coeff. of Disch. = 0.64		Volume passed into tank, cubic feet	Time, seconds	Volumes of Free Air in Tank	
			Cubic foot per second	Cubic feet per minute			Cubic foot per second	Cubic feet per minute
0.214	12	$8\frac{9}{16}$	0.184	11.0	17.49	96.5	0.181	10.9
0.214	19	$9\frac{9}{16}$	0.194	11.6	23.01	108.0	0.212	12.7
0.214	16	$7\frac{1}{16}$	0.167	10.0	16.11	91.5	0.171	10.3
0.214	16	$5\frac{7}{8}$	0.152	9.1	13.82	88.5	0.156	9.4
0.214	16	$5\frac{3}{4}$	0.151	9.1	13.82	88.0	0.157	9.4
0.441	9	$2\frac{1}{4}$	0.194	11.6	19.33	96.0	0.201	12.1
0.441	12	$3\frac{1}{8}$	0.237	14.2	20.48	86.0	0.238	14.3
0.441	13	$1\frac{15}{16}$	0.186	11.2	22.55	116.0	0.194	11.6
0.441	16	$4\frac{1}{8}$	0.271	16.3	22.86	87.0	0.263	15.8
0.441	16	$4\frac{1}{4}$	0.267	16.0	19.40	74.0	0.262	15.7
0.797	12	$\frac{15}{16}$	0.228	13.7	20.18	86.0	0.238	14.3
Averages			0.203	12.2			0.207	12.4

Another apparatus for measuring low-pressure air by means of a nozzle is shown in Figs. 2 and 3. It is recommended as standard practice by the Compressed Air Society of New York, in its 1930 bulletin.

The air from the compressor passes first into an ordinary receiver *A*, where the pressure is kept constant by bleeding the air through a globe valve into receiver B. The latter is 10 to 15 ft. long, and in diameter not less than $2\frac{1}{2}$ times the smallest diameter of the nozzle throat. These ample dimensions insure the absence of eddy currents and a low speed of flow through the receiver. In the right-hand end of receiver B is a standard nozzle C (Fig. 3), by which the air is discharged to the atmosphere. The nozzle has polished, curved inner surfaces; for its dimensions, see Table III. For any given case, the size of the nozzle is such that, while all the air delivered by the compressor is being discharged through it, the pressure in B is maintained at not more than $1\frac{1}{2}$ lb. per sq. in. above the atmosphere. Attached to receiver B is a water-column manometer D, from which the air pressure is read. About 2 ft. back of the nozzle is a thermometer E, for showing the temperature of the air in the receiver.

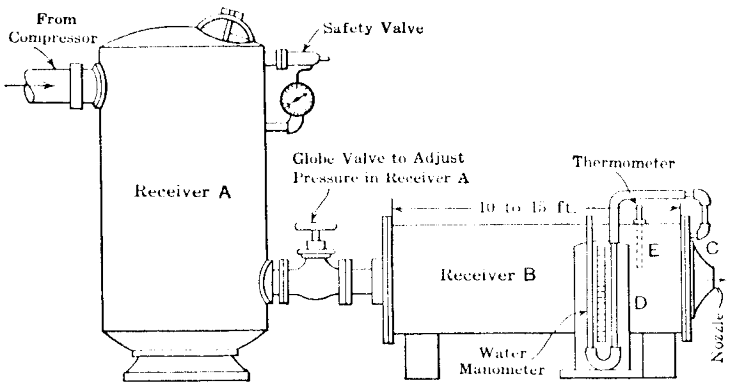

FIG. 2.—Apparatus for Making a Low-pressure Nozzle Test (Trade Standards, Compressed Air Society.)

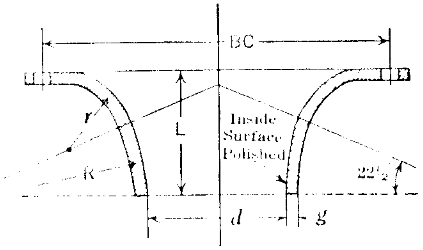

FIG. 3.—Standard Nozzle for the Measuring Apparatus in Fig. 2.

As the temperature of the air flowing through the nozzle generally ranges from 150° to 250° F., the volume discharged must be corrected back to the equivalent cubic feet at the temperature of the compressor intake. The necessary corrections are embodied in the following formula for computing the volume discharged, by Sanford A. Moss, of the General Electric Research Laboratory (see *Trans.* Am. Soc. Mech. Engs., Vol. 50, 1928):

$$q_3 = \frac{2.552D^2cT_3}{p_3}\sqrt{\frac{Bi}{T_1}},$$

TABLE III.—DIMENSIONS OF NOZZLE (FIG. 3).

	Dimensions, inches				Free Air Per Minute, Cubic Feet	
d	L	R	r	BC	Max.	Min.
2	1.70	2.8	1.0	$13\frac{1}{4}$	340	140
3	2.55	4.2	1.5	$13\frac{1}{4}$	755	290
4	3.40	5.6	2.0	$13\frac{1}{4}$	1,330	520
5	4.25	7.0	2.5	$13\frac{1}{4}$	2,065	810
7	5.95	9.8	3.5	17.0	3,990	1,560
9	7.65	12.6	4.5	$21\frac{1}{4}$	6,600	2,575
11	9.35	15.4	5.5	$27\frac{1}{4}$	9,860	3,850

Nozzle coef., 0.99. Capacity is figured on 20 in. maximum and 3 in. minimum of water column.

where q_3 = cubic feet of air per minute, at pressure p_3 and temperature T_3; that is, the conditions at the compressor intake;

D = minimum diameter of nozzle throat, inches;

c = coefficient of nozzle discharge, 0.98 to 0.99;

p_3 = absolute pressure, pounds per square inch, at which q_3 is to be expressed;

T_1 = absolute temperature Fahr. inside of nozzle;

T_3 = absolute temperature Fahr. at compressor intake;

i = difference between absolute pressures p_1 and p_2, lbs. per square inch, at up and downstream sides of nozzle; that is, the pressure drop through the nozzle, in inches of water column;

B = value of p_2 in inches of mercury (barometric reading), corrected for the temperature shown by the thermometer, and the barometer calibration constant.

Example. Nozzle, 5 in. diameter; air temperature on entering nozzle, 192° F.; compressor intake temperature, 69° F.; observed manometer reading, 10.4 in.; barometer, 29.83; intake pressure, atmosphere (14.65 lb.); nozzle coefficient, 0.99. Under these conditions, $T_1 = 192 + 460 = 652°$ and $T_3 = 69 + 460 = 529°$ F.

Substituting the above values in the formula:

$$q = \frac{2.552 \times 5^2 \times 0.99 \times 529}{14.65}\sqrt{\frac{29.83 \times 10.4}{652}} = 1{,}573 \text{ cu.ft. per min.}$$

Measurement of High-pressure Air.—(1) For comparatively small volumes, the air delivered by a compressor may be passed into a tank or receiver of known capacity, provided with a pressure gage; (2) the air may be measured by causing it to displace water in a tank; (3) the air may be metered.

Measurement of Discharge through Orifices, without Using a Second Receiver.—This method is applicable to both small and large volumes. The compressed air is discharged through orifices of known area, and the equivalent volume of free air computed.

One apparatus for this method consists of a short piece of wrought-iron pipe, in which are set a number of short branches, each provided with an orifice, the orifices being of different diameters.* Each branch has a valve, the whole resembling a manifold as used for connecting a group of rock drills to an air main. Fig. 4 shows a meter with eight orifices, ranging from $\frac{3}{32}$ to $\frac{5}{8}$ in. diameter. The orifices are accurately reamed holes in disks of steel plate, held by flanges in the outer ends of the branch pipes. For the larger orifices the disks are $\frac{1}{2}$ in. thick; for the smaller, $\frac{3}{8}$ in. The back or pressure side of each hole is rounded to a radius $\frac{1}{16}$ in. less than the thickness of the plate. The actual diameter of the finished hole is measured by a micrometer and the area calculated.

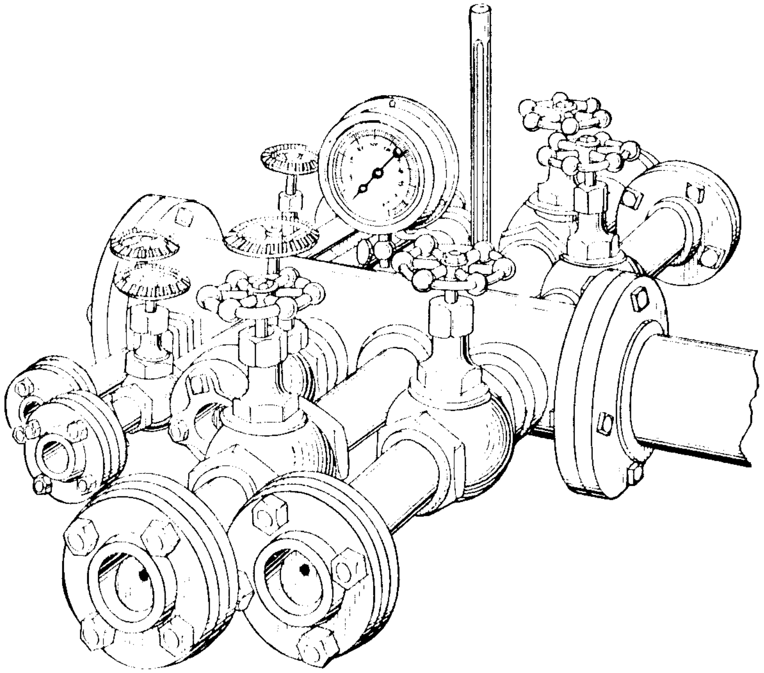

FIG. 4. Manifold and Nozzles for Measuring the Discharge of Compressed Air (*Engineering and Mining Journal*, May 25, 1912).

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* F. D. Holdsworth, Chief Air-Compressor Engineer, Claremont Plant, Sullivan Machinery Co. *Engineer and Mining Journal*, May 25, 1912, p. 1028.

The rate of flow through an orifice of this form is obtained from Fliegner's formula,

$$G = 0.53\frac{AP}{\sqrt{T}}$$

where
G = flow, pounds per second;
A = area of orifice, square inches;
P and T = absolute pressure and temperature (F.) of the air behind the orifice.

For the formula for the weight of a cubic foot of air, see the footnote under Table I.

Several small orifices are used instead of a single large one, because, though the formula is accurate for orifices not greater than $\frac{5}{8}$ in., it is inaccurate for larger ones. Also, by having a series of orifices of different diameters, any desired combination may be made according to the volume of air to be measured. Tapped into the main pipe of the manifold is an accurate pressure gage and a well for a thermometer, which, for a 2-stage compressor, should read to at least 300° F.

To prepare for a test, disconnect the air main near the receiver and attach the manifold. Stop all other outlets from the receiver, so that all the air entering the receiver will pass through the orifices. Place another thermometer in the compressor intake, close to the cylinder. The compressor speed is recorded by a revolution counter. Determine by trial the proper combination of orifices to maintain the desired pressure. Run the compressor about two hours, discharging through the orifices, so that both pressure and temperature will become maximum and constant.

One observer begins recording the compressor revolutions, and another, at 1-minute intervals, takes the temperature and pressure readings at the orifices; at the same time, the intake temperature is also recorded. In 10 or 15 minutes enough temperature and pressure readings will be obtained, and a final reading is made of the revolution counter, to find the total number of revolutions for the period of the test. The total piston displacement is then computed. Finally, the quantity of air discharged through the orifices, determined by the formula and divided by the total displacement, gives the volumetric efficiency.

For accurate results, obtain the local barometric reading from the nearest Weather Bureau station, or otherwise. This, with the temperature at the compressor intake, is used in the formula, $W = \frac{1.325B}{T}$

(see note under Table I), for reducing to cubic feet of free air per minute the pounds of air per second, computed by Fliegner's formula. In testing a stage compressor, only the displacement of the low-pressure cylinder is used, since this measures the volumetric capacity of the compressor.

A method of measuring air by Pitot tubes was adopted at a mine in northern Michigan, for finding the quantity of air used in a large blacksmith shop.* The compressor was working at a pressure of 75 to 80 lb.

Two Pitot tubes, made of copper tubing $\frac{1}{16}$ in. in outside diameter, were soldered into a $\frac{3}{8}$-in. to $\frac{1}{2}$-in. bushing. This bushing in turn was screwed into a hole tapped in the side of a piece of 1-in. iron pipe, 20 in. long, which was inserted in the air main. The Pitot tubes projected $\frac{1}{4}$ in. into the pipe; one of them was directed toward the air current, the other set at right angles to it. They were connected to a U-tube, made of two gage glasses joined at their lower ends by rubber hose. Either a water or a mercury gage may be used to determine the velocity head of the air current. The 20-in. piece of 1-in. pipe was provided with a thermometer and a pressure gage, set behind the Pitot tubes and at a little distance from them, to avoid producing eddies.

The volume of air flowing through the 1-in. pipe was computed by the formula

$$Q = 135\sqrt{\frac{H(P + 14.7)}{t + 460}},$$

where Q = cubic feet of free air per minute, at 60° F. and 28.5 in. barometer;

H = height of water column, inches, representing the velocity head (if mercury be used, multiply the reading by 13.6, which is the specific gravity of mercury);

P = reading of pressure gage;

t = thermometric temperature of the compressed air (deg. F.).

To furnish some storage capacity, and so prevent sharp peaks in the flow of air caused by the intermittent use of the drill sharpener and the shop hammer, the measuring device was set in the air line at a little distance from the shop. Measurements were made on a Saturday afternoon, when air was being used only for the shop and rockhouses. The result showed: total free air compressed, 670,000 cu.ft.; blow-off, 338,000 cu.ft.; line loss, 118,000 cu.ft.; blacksmith shop, 27,000 cu.ft.; rockhouses, 187,000 cu.ft. The high line loss was largely due to expansion and contraction in the spiral-riveted pipe and

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* B. B. Hood, *Engineering and Mining Journal*, June 27, 1914, p. 1283.

its joints. Most of the leakage was in the 500 ft. of air main nearest the compressor, where the variations of temperature were naturally greatest. Underground, the piping was found to be practically tight.

"Ready-made" Meters of several forms are on the market; for example, the "Drillometer" (New Jersey Meter Co., Plainfield, N. J.) the Clark meter, and the instrument of the Bailey Meter Co., Boston.

The Bailey meter measures the pressure difference across an orifice placed between a pair of flanges in the pipe line. A special orifice plate is used, made of $\frac{1}{32}$-in. Monell metal. The area of orifice is proportioned to the size of pipe, so that, at average rates of flow, a pressure-drop of about $\frac{1}{2}$ lb. takes place. The record is automatically made on a direct-reading chart, with uniform graduations. (For a detailed description, see *Met. and Chem. Eng'g.*, April 15, 1916, p. 456.)

The Clark meter operates on the principle of the volumetric displacement of a piston, the linear travel of which is recorded as the volume of air passing.

The Drillometer (Fig. 5) is a portable, direct-volume gage, employing multiple nozzles. It has but one moving part, the weighted piston a, in the metering cylinder b, and a small piston d in the oil dashpot c. The rod above a moves freely, without contact, inside of the sight glass f, and the height of its upper end is read on the accompanying scale, which records cubic feet of free air per minute.

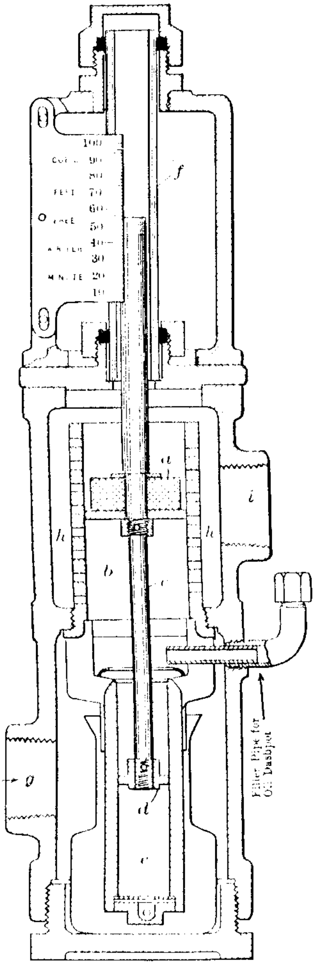

FIG. 5.—The "Drillometer," for Measuring Compressed Air.

At g air enters the space around the dashpot cylinder c, passing thence through ports into the metering cylinder b. Through the walls of cylinder b are drilled a large number of small holes, accurately spaced and reamed. In passing to the outlet chamber h, the compressed air lifts piston a, thus exposing some of the holes to the flow. The air leaves the meter at i.

The small "head" or difference of pressure (a few ounces per square inch), produced between the interior of cylinder b and the chamber h, is fixed by the relation of the weight of the moving element to the area of the piston a on which the difference of pressure acts. That is, the moving element rises until its weight is exactly supported by the difference of pressure; so that the pistons and rod float in static balance, in a position corresponding to the volume of air flowing and the number of holes exposed in cylinder b. The scale is calibrated by comparison with a standard instrument.

This meter is made in two sizes: the smaller for flows of 10 to 100 cu. ft., and the larger for 50 to 300 cu. ft. of free air per minute.

It may be suggested that the water meter made by the Worthington Pump & Machinery Co. might readily be modified for measuring compressed air. The measuring chamber of this meter contains a "wabbling disk," resting at its apex on a ball, and set between the inlet and discharge openings. A slit cut in one side of the disk fits a radial septum or division in the disk chamber; water is admitted on one side of the septum and discharged at the other side. The water pressure causes the disk to make a uniform wabbling motion, which passes water continuously through the meter; though there is at no time any free opening, through which water could flow without moving the disk. A short rod, attached at right angles to the upper side of the disk, communicates motion to a gear-train, which operates a counter registering cubic feet of water on a dial. This meter, like some others, is of the "inferential" type; that is, it must be rated (calibrated) to determine its discharge.

Measuring Tanks.—Two large vertical tanks are generally used, connected with the compressor and with a pump for supplying water to the tanks, long vertical gage glasses being provided for reading the water level. The air in the two tanks is always separated by a piston-like mass of water, which flows back and forth from one tank to the other. The change in level, multiplied by the area of the tank, is equal to the volume of air that has flowed out. In order that the observed differences of water level shall accurately represent corresponding volumes of air, it is evident that the internal cross-sections of the tanks must be constant throughout their heights.

The machine to be tested is connected to one tank, both tanks are filled with air at the desired pressure, and the pump keeps the air pressure constant by forcing in water. On starting the drill, or other machine, the height of water in the gage glass is noted. After a run of sufficient length to secure accuracy, the drill is stopped and the height of water in the gage glass on the tank which has been used is again noted. The increase in volume of water in the tank is the volume of compressed air used during the run; from this the volume of free air is computed. Having two tanks, one is used while the other is being filled with air, thus allowing tests to be run for any desired time.

Fig. 6 shows an apparatus, which though more elaborate is practical and convenient, and the tanks need not be of uniform cross-section throughout.* Instead of using long gage glasses, reaching from top to bottom, one tank has two short glasses, f and h, one at top, the other at the bottom; the companion tank has a single gage, at the top.

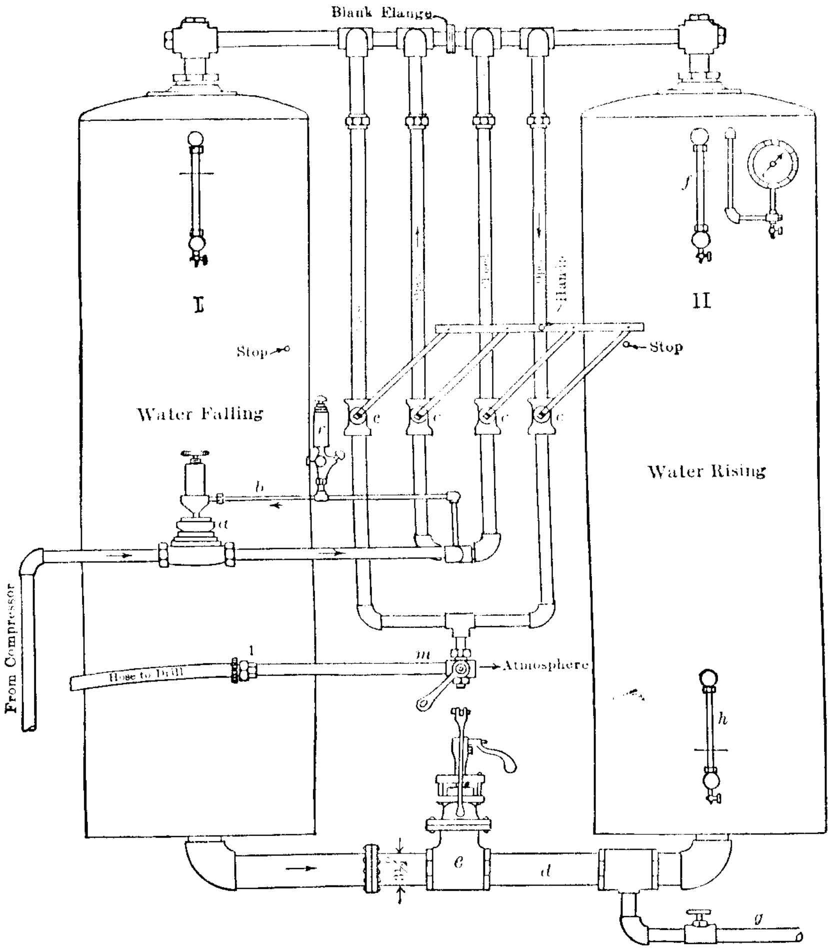

Fig. 6.—Measuring Tanks for Rock-drill Testing (W. S. Weeks, *Mining Press*, December 15, 1917).

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* Walter S. Weeks, *Mining Press*, December 15, 1917, p. 855.

As the tank air must be kept at constant pressure, a reducing valve is provided (the valve known as the "*S*-4 locomotive governor," made by the Westinghouse Air Brake Co., is recommended). It is arranged as at *a*, Fig. 6. The control pipe *b* should not be connected directly to either tank, as the valve continuously exhausts air to the atmosphere. A lubricator *r* is placed on pipe *b*, for feeding a thin oil to the valve.

To transfer the air from one tank to the other, a 4-way valve is generally used (see Fig. 7). It has the disadvantage of being liable to leakage. Fig. 6 shows an arrangement of four pipes, each provided with a single, 1-in. cut-out valve *c*. The valve levers are connected, so that all are thrown together, thus producing the effect of a four-way valve. In pipe *d*, connecting the tanks at the bottom, is a quick-closing gate-valve *e*.

The volume of one tank only needs to be measured, that having the two gage glasses. First place a mark on glass *f*, and fill the tank with water to that point through pipe *g*. Close valve *e*, and draw off water through pipe *g* until the water level is sighted on glass *h*. Then carefully draw off more water, until a convenient even number of cubic feet is reached, and mark the corresponding point on glass *h*. Thus, when in operation the water rises from the lower to the higher mark, the measured volume of air has been displaced.

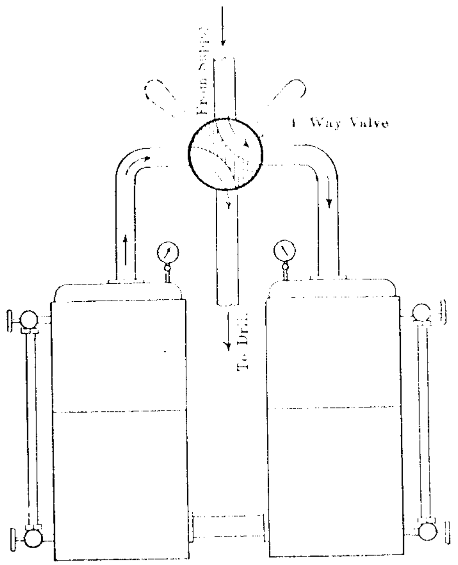

FIG. 7. Tanks for Measuring Air Consumption of Rock Drills.

To test a drill, connect the hose at *l*, as shown; then fill tank I with water; close the gate-valve *e*; bring the water level in tank II to the lower mark; open the gate-valve and throw valves *c* to the position shown, thus forcing water from tank I to tank II until it reaches the upper mark; then close the gate-valve. The reducing valve *a* is adjusted to hold the pressure constant, notwithstanding the small changes in hydraulic head on the air. When necessary, the air in the tanks may be exhausted to the atmosphere through the three-way valve *m*. The drill test may be continued as long as desired by running the water a number of times from one tank to the other. For ordinary testing, the volume between the gage-glass marks on tank II should be 16 or 18 cu. ft.

A less expensive apparatus consists of two small tanks, say 2 ft. in diameter by 4 or 5 ft. high, connected at the bottom by a 3-in. pipe and provided with gage-glasses (see Fig. 7).* The tanks are about half filled with water. Pipes from the top of each tank are connected to a common four-way valve. With the valve in the position shown in the cut, water in the right-hand tank is forced by the air pressure into the left-hand tank, and the air in the latter is driven out through the valve to the drill. When the water level reaches a definite point near the upper end of the gage-glass on the left-hand tank, the valve handle is quickly thrown to the position shown by dotted lines. The process is thus reversed, the air in the right-hand tank then passing to the drill.

The total volume of compressed air used is found by counting the number of times each tank is filled and emptied during a given time, and the corresponding volume of free air is computed as already explained.

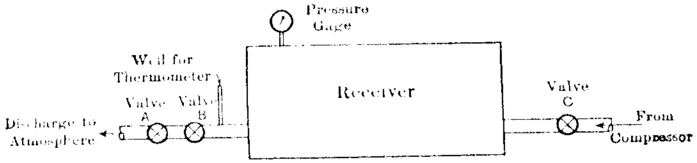

FIG. 8. Air Receiver Arranged for Measuring Air from a Compressor.

Measuring Air Discharge with One Tank.†—An accurate method is as follows. To one end of the air receiver is connected the air pipe from the compressor. At the other end is a horizontal discharge pipe, provided with two globe valves set close together and a well for a thermometer (Fig. 8). This pipe may be a part of the air main, disconnected from the service line so as to discharge into the atmosphere. On the receiver is an accurately calibrated pressure gage.

To begin a test, first open valves *A* and *B* wide; then, with the compressor at normal speed, gradually close valve *B* until the tank pressure is constant. After a short run, to insure that pressure and temperature conditions have become constant, close valve *A* tight and allow the tank pressure to rise to 10 or 15 lb. above normal working pressure. Then close valve *C*, stop the compressor, open valve *A* quickly, and o exact simultaneous readings of time and pressure, as the tank pressure falls due to escape of air through valve B. An assistant holding a watch should call " read " at uniform intervals of 10 or 15 seconds, continuing until the pressure has dropped considerably below the working pressure. Finally, plot the readings on cross-section paper, to obtain a time-pressure curve (Fig. 9).

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* George H. Gilman, Sullivan Machinery Co., *Compressed Air Magazine*, Aug. 1912, p. 6510.

† E. E. Fessenden, *Compressed Air Magazine*, February, 1913, p. 6714.

From Chapter II, p. 10, if the weight of the volume of compressed air in the tank = W, then $PV = WRT$, whence $W = \frac{PV}{RT}$. W is taken as the entire volume of the tank and its connections between valves B and C; it is determined by weighing the volume of water required to fill the tank and connections, and dividing by 62.355 (weight of 1 cu. ft. of water at 62° F.).

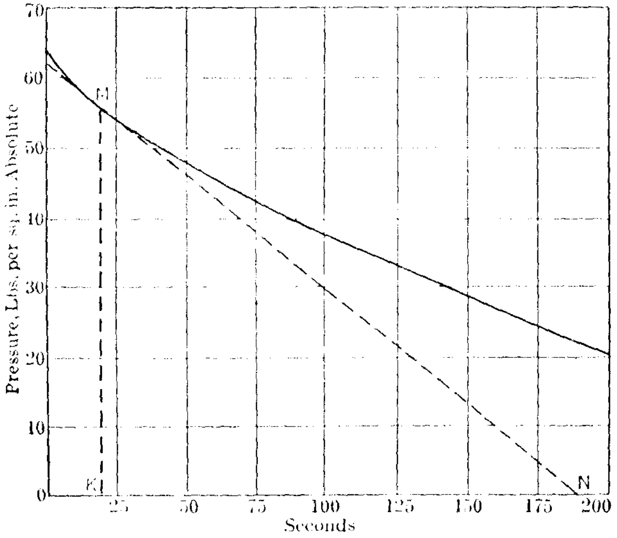

Fig. 9.—Time-pressure Curve.

Differentiating the above equation with respect to time: $\frac{dW}{dt} = \frac{V}{RT} \times \frac{dP}{dt}$, in which $\frac{dW}{dt}$ is the rate of decrease of weight of air in the tank, or the rate of discharge in pounds per second, while $\frac{dP}{dt}$ is the rate of decrease of pressure.

In Fig. 9, M is a point on the time-pressure curve corresponding to the normal working pressure. MN is carefully drawn tangent to the curve at M, and MK is parallel to the pressure axis. Then $\frac{dP}{dt} = \frac{MK}{KN}$. Substituting in the differential equation: $\frac{dW}{dt} = \frac{V}{RT} \times \frac{MK}{KN}$. Reading from the diagram the values of MK and KN, the rate of discharge, in pounds per second, is found directly. As the weight of air discharged equals the weight drawn into the compressor, the free air capacity is found from $V_0 = \frac{WRT_0}{P_0}$, where V_0 is the volume, in cubic feet of air per second, at the intake pressure P_0 (pounds per square foot absolute) and at the absolute temperature T_0, and W is the weight of air discharged per second as already found.

Another mode of using a single tank, especially adapted to testing rock-drills, is shown in Fig. 10. A vertical tank or receiver is connected to the main air line from the compressor by a 1-in. pipe, another 1-in. pipe leading to the rock-drill. On the receiver is a line of overlapping water-gages, or a single long gage glass, extending from bottom to top. Connected to the bottom of the receiver is a 3-in. pipe from a pump (or a water tank at an elevation sufficient to give a static pressure equal to the air pressure).

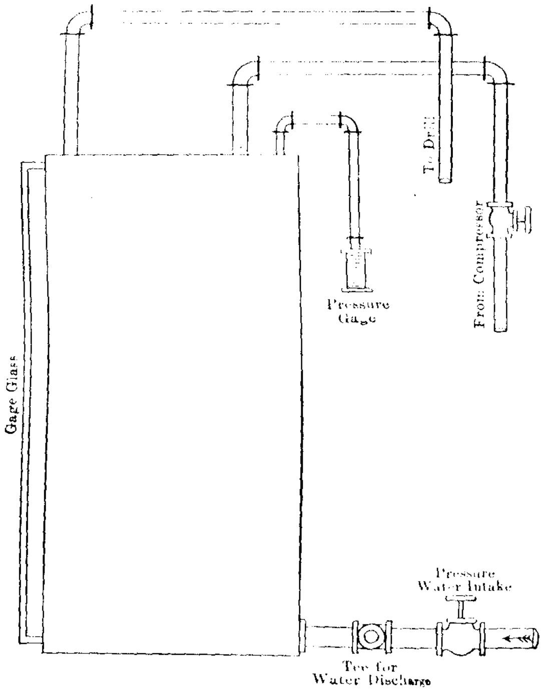

Fig. 10.—Diagram of Tank for Measuring Air Consumption of Rock-drills.

To begin a test, run in enough water to show at the bottom of the gage glass, and make a chalk mark opposite this point. Next, turn on the compressed air until the receiver is filled at the desired pressure; then turn off the air and start the drill. During the run, maintain a constant-pressure in the receiver, as indicated on the pressure gage, by regulating the valve in the water pipe. When the drill is stopped, mark instantly the height of water level on the gage glass by another chalk mark. Note the distance between the marks, and the elapsed drilling time. The cross-section of the receiver being known, compute the volume of compressed air used, and the corresponding volume of free air. From the depth of hole drilled the air consumption per linear inch or foot of hole may also be found.

If it is be assumed that the receiver pressure is constant, the volume V of free air used is

$$V = RA\left(\frac{P + H}{H}\right)$$

where R = rise of water level, feet;
A = cross-section of receiver, square feet;
P = initial gage pressure, pounds;
H = atmospheric pressure.

In case the receiver pressure decreases from P to P' during the test, let V = initial volume of free air in the receiver at pressure P, and V' = final volume of free air at pressure P'. Then

$$V = AB\left(\frac{P + H}{H}\right) \text{ and } V' = A(B - R)\left(\frac{P' + H}{H}\right),$$

where A, R and H are as above, and B = total height from initial water level to top of receiver, feet. Then the volume of free air used $= V - V'$.

MEASUREMENT OF LARGE VOLUMES OF COMPRESSED AIR

Venturi Tube.—This old device for measuring the flow of water has also been used with some success for air. It is not accurate, however, especially for small volumes, and, because the flow through the tube varies as the square root of the pressure difference, the flow scale is inconvenient. The Venturi meter is too well known to require discussion here.

Metering Compressed Air at the Gold Mines of the Rand, South Africa.—The very large quantity of air used in this district is indicated by the fact that, at the end of 1924, the horsepower of the air-compressing plants aggregated about 300,000. Consequently, the question of accurate measurement has for some years occupied the attention of the best mechanical engineers of the Transvaal gold fields. In a valuable paper presented Jan., 1925, before the South African Institute of Engineers, E. J. Laschinger, M.E., discusses the present status of compressed-air measurement on the Rand, and describes at some length the development of a new meter—the "F. L. M. Graphic Flow Recorder"—about 300 of which are now being used at the different mines of the district. They were intended primarily as distribution meters for the mines of the Central Mining and Rand Mines Group, to assess the quantity of compressed air used in various services or sections of the mines operated under that administration.

The principal desiderata for meters for such service are: (1) Reasonable accuracy over the whole range of measurement; (2) Strength and simplicity, to withstand rough handling and the air concussions due to blasting underground; (3) Reliability, so as not to be easily choked by air carrying water, oil and dust; (4) They must not only indicate momentary flow, but also give a graphic flow time-record, for comparison with the work done by the air in the corresponding mine or section of workings; (5) They must not be sluggish, and when no air is passing should instantly indicate zero; (6) They should have a long life, require little attention and be dependable.

The F. L. M. meter is of the "weighted-gate" type, the gate controlling a differential orifice or flow opening, in such manner that delicacy of movement of the gate with small flows is combined with relative accuracy of the orifice with large flows. The controlling factor is the weight of the gate, and the theory of the meter is to make equal angular movements of the gate correspond to equal increments of flow.

Two forms of this meter have been designed by Mr. Laschinger, in collaboration with Messrs. Furnivaal and Maclaren of the Rand Mines Air Testing Department. Fig. 11 shows the "shield" type, in which the size of orifice depends upon the varying distance between the trailing edge of the gate and the curved shield. Fig. 12 shows the "slotted-plate" type, in which the shield is circular, conforming to the movement of the trailing edge of the gate; in the shield is a slot of diminishing width as the gate opens, through which the air passes.

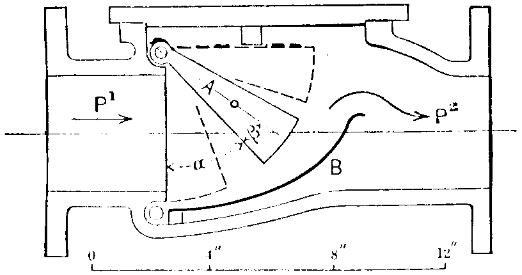

FIG. 11.—Shield Type of Air Meter.

After much experimental work and practical service underground, the type of meter shown by Fig. 11 was preferred. In Mr. Laschinger's paper, cited above, is given the mathematical theory of the design. A gate A, of substantial weight, hangs vertically over the opening in the body of the main casting. Below the lower edge of the gate is a curved brass shield B, which is quite close to the edge of the gate, when the gate is shut, but is progressively farther away as the gate swings open. That is, the gate's angular movement corresponds to the rate of flow of air, and equal increments of movement represent exactly equal volumes of flow. An arm attached to the outer end of the spindle of the gate actuates a cam, mounted on a separate spindle, and on the cam is another arm carrying a pointer and a pen. The pointer indicates the rate of air flow on an equally divided scale, and the pen marks a continuous record on a chart wrapped around a clock drum. To minimize fluctuations a "dampening vane," working in an oil chamber filled with thick cylinder oil, is attached to the cam. The chart is 12 in. long by 5 in. wide. Since the flow scale is uniform, a planimeter may be run over the record, to obtain total and (or) average rate of air consumption. In one mine, having 30 distribution meters of all sizes on different levels and sections, the total readings for one month have agreed at from 2 to 6 per cent less than the total air used (40,000 cu. ft. per min.), as measured in bulk at the surface.

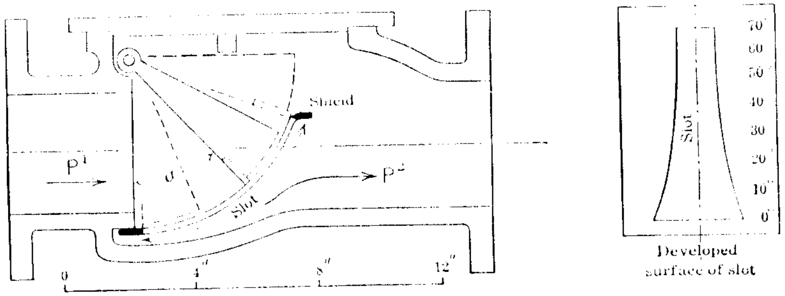

Fig. 12.—Slotted-plate Type of Air Meter.

The meters are calibrated for a standard gage pressure of 80 lb. and a temperature of 70° F., which are taken as normal for the existing conditions. Fig. 13 is a correcting chart, for converting any other pressure or temperature to the normal basis. These corrections are generally small. For applying them a chart is preferable to automatic devices, which introduce too many complications. A 6-in. meter indicates to within 1 per cent plus or minus, over a range from full capacity to 1/20 of full load, with a maximum drop of 1 lb. per sq. in. at full load.

Instructions for using the chart (Fig. 13). For any other pressure and temperature than the standard (80 lb. gage and 70° F.), the following directions apply:

Example. Rate of flow shown by the meter = 2,800 cu.ft. free air per minute; air pressure, 50 lb.; air temperature, 140°. To obtain correct rate of flow, read along line marked 2,800 on left-hand side of chart. At intersection of this line with pressure curve marked 50 lb., the value of 2,300 cu.ft. free air is shown on bottom scale; this is the correct flow at 50 lb. To correct for temperature, read along line 140° at right-hand side. At intersection of this line with the temperature curve, the value 0.94 is shown. Hence, 2,300 × 0.94 = 2,162 cu.ft. free air per minute, which is the correct amount passing through the meter.

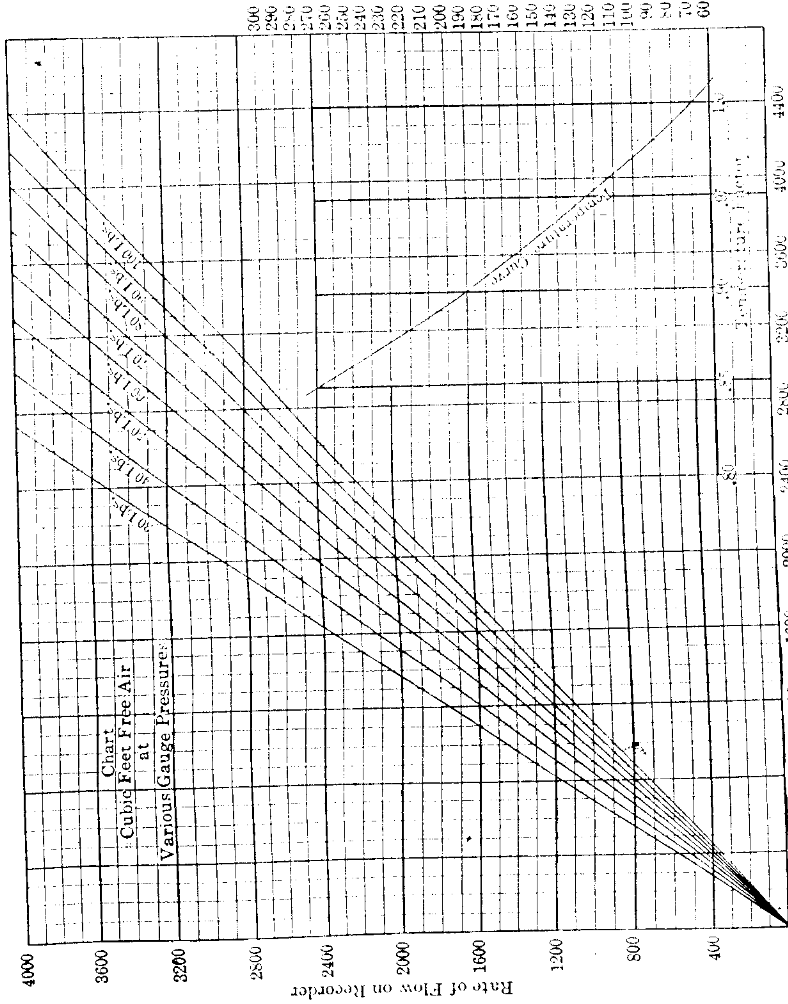

Fig. 13.—Correcting Chart, for Use with the I.L.M. Meter.

The dimensions indicated on Fig. 11 are for a 4-in. meter, for a maximum flow of 100 lb. air per minute (= 1,600 cu. ft.), at 90 lb. initial gage pressure. The maximum movement of the gate is through an arc of 70°. The gate weighs 8.34 lb.; angle of lead (β) is 13°; maximum drop of pressure across the orifice, 0.387 lb. per sq. in. (= 10.7 in. of water), and a differential pressure of 2.43 in. of water opens the gate at minimum flow.

In the F. L. M. meter the path of the air is direct, instead of turning through sharp angles, as is the case with many meters of other designs. Hence, loss of pressure by friction is minimized, and foreign substances are readily blown through. Several of the first of these meters, installed about 1922, have been found by test to be unaltered, simply needing cleaning. All parts are of gun-metal or brass, to resist corrosion. The exterior parts are inclosed in a locked case, with glass front for convenience of observation. In installing the meter, it must be set truly level, and connected up with a length of at least 5 ft. of straight air pipe on each side of the meter; all joints being made so that no jointing material protrudes into the pipe.

Effect of Pulsating Flow on Air Meters. Most of the air used in the Rand mines mentioned above, and measured by the F. L. M. meters, is compressed by turbo-compressors, of the Victoria Falls and Transvaal Power Co.,* and its flow is therefore non-pulsating.

Mr. Laschinger, in his paper, calls attention to the fact that the practical difficulties of measuring air having a pulsating flow are always great, and have not yet been overcome, notwithstanding that in recent years much experimental work has been done. Where there are violent pulsations, as is the case with reciprocating compressors, the F. L. M. meter is not to be recommended for accurate recording, unless excessive variations are dampened. There are two kinds of waves, that may be called pressure waves and velocity waves. The former travel with the velocity of sound and are governed by the same law; the latter, which produce a real pulsating flow, present a serious problem and cause peculiar phenomena in manometer measurements. It is known that the effects of the velocity waves can be reduced by providing, in the transmission main, capacities and restrictions—as receivers, orifices and other devices—for absorbing the kinetic energy of these waves. Mr. Laschinger and his colleagues have been experimenting with the purpose of discovering some practicable method of dampening down, if not eliminating, the effects of pulsating flow; hoping to find some device which, by attaching to any kind of air meter, will permit reliable registration.

A meter operating on the shunt principle has already been evolved, and is being used successfully in three of the Witwatersrand mines, for measuring main outputs at compressor stations. In a flange joint of the air main is inserted a diaphragm containing an orifice. Set in the main, and connecting the two sides of the orifice, is a small pipe, with a sensitive gate recorder. By establishing the relation between the main and shunted flows (similar to the principle of most electricity meters), the meter diagram is graduated to read the total flow. This device is suitable as a large station meter and its cost is moderate. It has the advantage of automatically dampening the pulsations of the main flow, as affecting the shunt flow; that is, the pulsations on the recording meter are very small as compared with those in the main pipe.

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* See p. 97, Chapter III of this book.

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